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Technical Summary Report

for

**Study to Evaluate the Feasibility of a
Feedback Controlled Variable
Conductance Heat Pipe**

Contract NAS2-5722

September 1970

Prepared for

Ames Research Center
National Aeronautics and Space Administration
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TECHNICAL SUMMARY REPORT

for

**STUDY TO EVALUATE THE FEASIBILITY OF A FEEDBACK
CONTROLLED VARIABLE CONDUCTANCE HEAT PIPE**

September 1970

Contract No. : NAS2-5722

Ames Research Center

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ABSTRACT

An evaluation of the feasibility of feedback controlled variable conductance heat pipes has been completed. Both active and passive control mechanisms were considered. A bellows system is used to provide the feedback control in the passive system while an electronic controller and heated storage volume are used in the active case. The following conclusions are based on the results of the study:

- **Both active and passive feedback controlled variable conductance heat pipes are feasible.**
- **Feedback controlled heat pipes will show a marked improvement in heat source temperature control over conventional thermal control heat pipes.**
- **Both the active and passive systems can be made stable.**
- **The greatest benefits of feedback controlled systems are realized in applications where high heat source resistances are present or where large variations in heat load or environmental conditions occur.**
- **An active feedback controlled system will give better temperature control than an equivalent passive system. Also, mass diffusion and varying sink conditions will be less detrimental in an active system.**

Preliminary designs have been completed for both active and passive feedback controlled variable conductance heat pipe systems. In general, an active system appears to have greater design flexibility while at the same time giving sharper temperature control than an equivalent passive system. It is, therefore, recommended that a prototype active feedback controlled heat pipe be fabricated and tested.

FOREWORD

This technical report describes the work completed under Contract NAS2-5722, "Study to Evaluate the Feasibility of a Feedback Controlled Variable Conductance Heat Pipe." The work was administered by Ames Research Center, Moffett Field, California. Mr. John Kirkpatrick was the NASA technical monitor.

The investigation was conducted by Dynatherm Corporation, Cockeysville, Maryland, under the direction of Dr. Walter Bienert, Program Manager. Mr. Patrick J. Brennan was the principal investigator.

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NOMENCLATURE

<u>Symbol</u>	<u>Definition</u>
A	Area
G	Solar flux constant
K	Effective thermal conductance between equipment shelf and panel
k	Thermal conductivity, spring constant
m	Mass
p	Pressure
Q	Heat load
R	Gas constant
R_s	Thermal impedance between the heat source and the vapor
S_i	Variable conductance control parameters
T	Temperature
t	Honeycomb panel skin thickness
V	Volume
y	Location of gas-vapor interface
α	Solar absorptance
β	Coefficient of volumetric expansion
ϵ	Hemispherical emittance
λ	Bellows displacement
η_f	Fin efficiency
χ	Coefficient of isothermal expansion
π	Vapor pressure in inactive part of condenser or in storage volume
σ	Stefan-Boltzmann Constant

Subscripts

a	Auxiliary fluid
c	Condenser
e	Evaporator
g	Non-condensable gas
H	High power condition
L	Low power condition

Subscripts**Definition**

o	Sink condition
s	Heat source
st	Storage volume
v	Vapor

Section 1

INTRODUCTION

The principle of a variable conductance heat pipe has been known and understood for years.^{1, 2, 3, 4} Serious efforts are now being made to apply the principle to various space-craft temperature control problems.⁵ In the conventional form, the variable conductance heat pipe is able to maintain its own temperature nearly constant while heat input or environmental conditions are changed.⁶ If the thermal impedance between heat source and heat pipe is small, the source temperature will also remain nearly constant. However, frequently this impedance is non-negligible and the source temperature will vary with varying heat input even if the heat pipe temperature is maintained at a constant level. A second difficulty with a conventional variable conductance heat pipe is its sensitivity to fluctuations of the sink temperature. Variable sink conditions may affect the temperature of the non-condensing gas which, in turn, can result in large excursions of the source temperature. Finally, there is the possibility of mass diffusion of the working vapor into the gas volume which would also result in large excursions of the source temperature in a conventional thermal control heat pipe system.

The variations in source temperature associated with the above problems can be reduced significantly by the use of a feedback system. The purpose of this contract was to study the feasibility of feedback controlled heat pipes. In principle, the feedback mechanisms being evaluated in this study monitor the source temperature and adjust the gas storage volume. As in the case of conventional thermal control heat pipes, a non-condensing gas is employed to control the heat rejection area of the heat pipe; but now the storage volume is variable and linked to the heat source. Both active and passive methods of control have been considered. In the passive system, the source

temperature is monitored by a thermal sensor which in turn adjusts the conductance of the heat pipe through a bellows system. In the active system, the non-condensable gas volume and therefore the conductance is adjusted by varying the partial pressure of the working fluid within the storage volume. The heat source is monitored electrically, and the signal drives a small heater inside the storage volume which in turn controls the temperature of the saturated working fluid. The use of feedback permits the heat source temperature to be monitored directly. As a result, the effect of changes in heat load, environmental conditions, etc., on source temperature are attenuated.

Efforts expanded under this contract have resulted in an analytic evaluation of feedback controlled heat pipes in general. In addition the influence of various physical parameters as they affect the temperature control afforded by either active or passive systems was determined. Based on these results a detailed design was generated for each system and their performance was calculated. Finally, a potential flight experiment which would include a feedback controlled heat pipe along with other thermal control devices was defined and analyzed. A complete description of the work accomplished is given in the following sections.

Section 2

GENERAL EVALUATION OF FEEDBACK CONTROLLED VARIABLE CONDUCTANCE HEAT PIPES

Following initiation of the contract, a general steady state analysis of a feedback controlled variable conductance heat pipe was performed. The model describes in differential form the variation of the heat source temperature T_s as affected by changes in the heat input Q , the heat sink temperature T_o , and other independent variables. The model is general in the sense that it is not limited to a specific configuration and treats various methods for obtaining thermal control in functional form. The only limitation imposed upon the system is that a noncondensing gas is used to control the available heat rejection area.

A system of five linear differential equations was required to define the model. These equations are presented in functional form along with related auxiliary equations in the Appendix. The actual application of these equations to a specific passive feedback controlled system is also presented in the Appendix in order to demonstrate their use. A matrix inversion was employed to obtain the desired solution which expresses the variation of T_s in terms of a number of independent variables along with functional parameters of the system.

$$dT_s = \frac{\left(\frac{\partial T_v}{\partial Q} + R_s (1 + S + S_1) \right) dQ + \frac{\partial T_v}{\partial T_o} dT_o + \frac{S}{\Psi_v} \left(\frac{dm_v}{m_v} + \frac{dT_g}{T_g} \right)}{1 + S + S_1 + S_2} \quad (2-1)$$

In deriving this equation, only those variables and parameters which afford a practical means of control or which could adversely affect it were considered. Methods such as either active or passive injection of the noncondensable gas, etc., have been neglected.

Equation (2-1) describes the change in source temperature as affected by changes in the heat load (Q), sink temperature (T_o), gas temperature (T_g), and mass of working vapor in the storage volume (m_v). Variations in the Q and T_o are associated with the performance requirements of the system (duty cycle, orbit, etc.). Uncontrolled changes in the gas temperature could result due to changes in the heat sink environment. Also, changes in the mass of working vapor present in the storage volume will result from mass diffusion between the working vapor and the noncondensable (diffuse interface) and will always have a tendency to occur. The effect of these variations must be minimized in order to prevent large variations in the source temperature. In a passive system, changes in Q , T_o , etc., are attenuated by the parameters S_i . In the active system, the mass of vapor in the storage volume is controlled by an electronic feedback unit. In this way the effect of variations in m_v is used essentially to negate those variations in source temperature that arise due to changes in heat load, etc.

Active control of the mass of vapor in the storage volume is affected by changing the saturation temperature of working fluid in the storage volume by a heater control. This is an active way of essentially varying the volume available for the noncondensable. A small amount of heat addition or removal to an insulated storage volume will result in substantial changes in its temperature. If a continuous wick extends from the heat pipe into the storage volume, saturated working fluid will be present, and these temperature changes will result in large changes of the vapor pressure. This, in turn, results in appreciable changes in the volume occupied by the saturated vapor thereby causing an equivalent change in the volume available for the noncondensable. By actively controlling the heat input so that proper displacement of the noncondensable occurs sharp control of the source temperature can be maintained.

The quantities S_i are parameters associated with passive control which is again accomplished by a variation in the volume occupied by the noncondensing gas. However, in the passive system, a variable storage volume is affected by actual physical displacement of a bellows system. Each of these parameters has a physical interpretation. S describes the effect of using a noncondensing gas and a fixed storage volume, as in conventional thermal control heat pipes. S_1 applies when a variable storage volume such as a bellows whose displacement is affected only by the vapor pressure (and therefore the vapor temperature) is used to contain the noncondensable. Finally, S_2 is associated with feedback control and is applicable when an auxiliary fluid which senses the source temperature is used to provide the driving force to vary the storage volume.

A detailed discussion of these parameters will be given in the next section. However, the effect of each of these parameters in controlling the source temperature can readily be determined from Eq. (2-1) and by considering their effect on the vapor temperature. The source and vapor temperatures are related through

$$\frac{dT_s}{dQ} = \frac{dT_v}{dQ} + R_s \quad (2-2)$$

where R_s is the thermal resistance ($^{\circ}\text{F}/\text{W}$) between the heat source and the vapor and is a constant. If feedback is not employed $\frac{dT_v}{dQ} \geq 0$, and thus in the limit $\frac{dT_s}{dQ} = R_s$.

Combining Eqs. (2-1) and (2-2) it follows that

$$\frac{dT_v}{dQ} = \frac{\left(\frac{\partial T_v}{\partial Q} - R_s S_2 \right) dQ + \frac{\partial T_v}{\partial T_o} dT_o + \frac{S}{\Psi_v} \left(\frac{dT_g}{T_g} + \frac{dm_v}{m_v} \right)}{(1 + S + S_1 + S_2)} \quad (2-3)$$

Therefore, in situations where variations in Q dominate ($dT_o, \frac{dT_g}{T_g}, \frac{dm_v}{m_v} \sim 0$), it is possible to have $\frac{dT_v}{dQ} = -R_s$ and thus $\frac{dT_s}{dQ} = 0$ by having a large S_2 relative

to $(1 + S + S_1)$. In this instance, since S is always greater than zero, it is best to design a configuration for which S_1 will be negative and S will be small in order to have $(1 + S + S_1)$ approach zero. With this sort of configuration it is possible to even have $\frac{dT_s}{dQ} < 0$. For cases where large changes in sink temperature are predominant it is desirable to have S_1 greater than zero and the sum of the S_i as large as practical considerations will permit. Finally, in practical applications variations in the gas temperature may occur, and it is also possible to have diffusion of the working vapor into the volume occupied by the non-condensable (dm_v). Either of these effects could cause large variations in the source temperature in a conventional thermal control heat pipe. However, by implementing the system with a variable storage volume and feedback, such that $(1 + S_1 + S_2)$ is large compared to S , large attenuation of these effects can be achieved.

In most applications, changes in each of the independent variables (Q , T_o , T_g and m_v) will probably occur. From the previous discussion, it can be seen that an optimum passive control system is difficult to define when each of these effects are present. However, in general, it can be said that the use of feedback ($S_2 > 0$) will always result in improved temperature control over an equivalent conventional system. Also, the larger the magnitude of the feedback parameter, the better the control. As regards an active feedback controlled system, S_2 and S_1 will be zero; however, because a storage volume is still required the parameter S will still be present. Thus in an active system, provided that large variations in the gas temperature do not occur, it is desirable to have S sufficiently large so that $S/(S + 1)$ approaches one (1). This permits the maximum effect of feedback controlled variations of the mass of the vapor (m_v) to be realized. At the same time, because of the nature of the system, appreciable passive attenuation of variations in heat load and sink temperature will be realized. As regards the deleterious

effects that will result if large variations in gas temperature occur, the use of a heated storage volume and the fact that only small variations in its temperature are required to give proper control will substantially reduce this variation in an active system.

Section 3

PASSIVE FEEDBACK CONTROLLED VARIABLE CONDUCTANCE HEAT PIPE SYSTEM

A passive feedback controlled variable conductance heat pipe is shown schematically in Figure 3-1. The control system consists of two bellows, and a sensing bulb located near the heat source. The inner bellows contains an auxiliary fluid and is connected to the sensing bulb by a capillary tube. As will be demonstrated later, it is best to use an incompressible liquid as the auxiliary fluid. Variations in the source temperature will cause a change in the pressure of the auxiliary fluid which results in displacement of the inner bellows and therefore of the larger outer bellows. The outer bellows is used as a variable storage volume for the non-condensable. Thus, by relating the displacement of the bellows system and therefore of the vapor-gas interface in the heat pipe to the heat source, a feedback controlled variable conductance system which regulates the source temperature is affected.

A. Evaluation of Control Parameters

The degree of control that is afforded by a passive system is determined by the magnitude of the parameters S_1 , which were discussed in the previous section. In order to define design parameters for the bellows system as well as select an auxiliary fluid it is necessary to study these parameters in more detail. Their general expressions are as follows:

$$S = \frac{\partial T_v}{\partial y} \frac{\partial y}{\partial V_g} V_g \psi_v \quad (3-1)$$

$$S_1 = - \frac{\partial T_v}{\partial y} \frac{\partial y}{\partial \eta} \frac{\partial \eta}{\partial p_v} p_v \psi_v \gamma \quad (3-2)$$

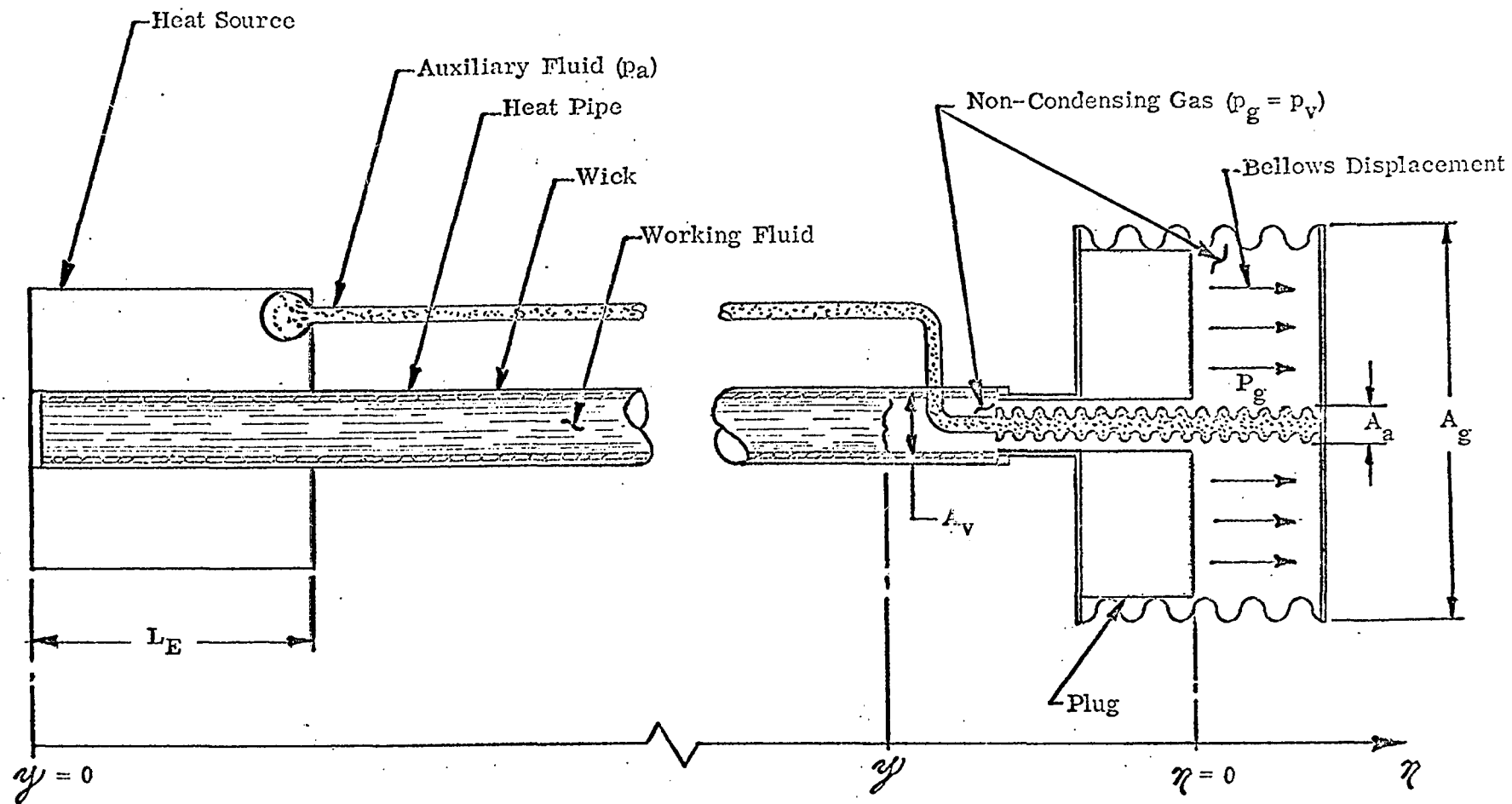


FIGURE 3-1
PASSIVE FEEDBACK THERMAL CONTROL HEAT PIPE SYSTEM

$$S_2 = \frac{-\partial T_v}{\partial y} \frac{\partial y}{\partial \eta} \frac{\partial \eta}{\partial p_a} \beta_1 \gamma \quad (3-3)$$

with

$$\psi = \frac{\partial \ln p}{\partial T}, \quad T = \text{Temperature}, \quad p = \text{Pressure}$$

$$\gamma = \left(1 - \beta_2 \frac{\partial \eta}{\partial p_a} \right)^{-1}$$

β_i = Parameters related to the kind of auxiliary fluid which is used and are defined as:

	Incompressible Liquid	Saturated Vapor
$\beta_1 =$	β/χ	$p_a \psi_a$
$\beta_2 =$	$\frac{-A_a}{\chi V_a}$	0

β = Coefficient of volumetric expansion

χ = Coefficient of isothermal expansion

Reference to Figure 3-1 and the above equations will give some insight into the factors affecting these parameters. The term $(\frac{\partial T_v}{\partial y})$ is associated with the variable conductance and relates to the dependence of the vapor temperature on the location of the gas-vapor interface. Each of the factors $(\frac{\partial y}{\partial V_g})$ and $(\frac{\partial y}{\partial \eta})$ are inversely proportional to the cross-sectional area of the vapor space in the heat pipe and reflect the change in the location of the gas-vapor interface with a change in the volume occupied by the gas. The product $(p \frac{\partial \eta}{\partial p})$ relates to the displacement of the storage volume associated with changes in either the vapor pressure or the pressure of the auxiliary fluid. The product $(p \frac{\partial \eta}{\partial p} \frac{\partial y}{\partial \eta})$ reflects the volume of gas in the storage volume associated with this displacement. The parameters ψ and β_1 are a measure of the sensitivity of the driving force to changes in temperature.

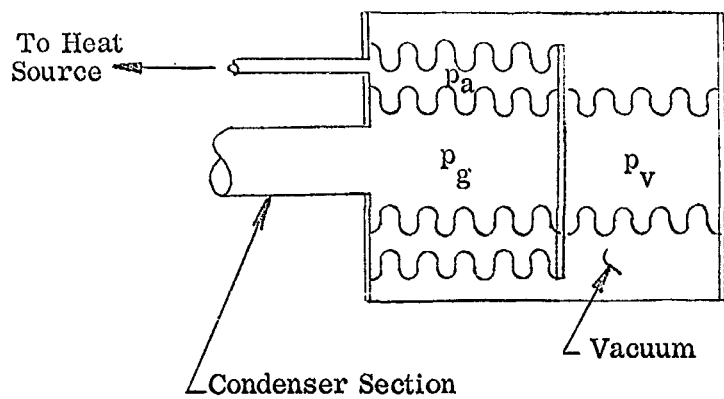


FIGURE 3-2a ($S_1 = 0$)

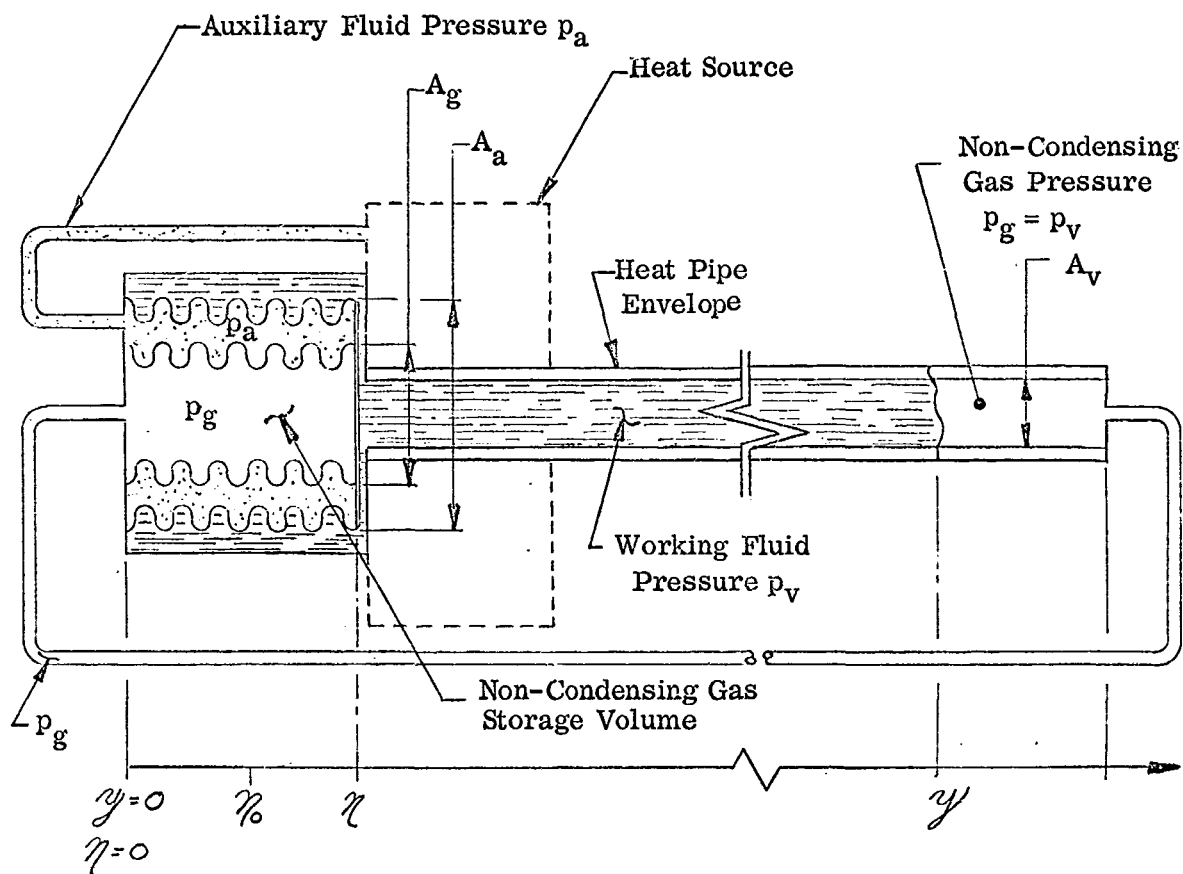


FIGURE 3-2b ($S_1 < 0$)

FIGURE 3-2

ALTERNATE CONTROL UNITS FOR PASSIVE FEEDBACK CONTROL

Finally, the parameter γ relates to the attenuation of driving force that results due to displacement of the inner bellows (increase in volume occupied by auxiliary fluid) when the auxiliary fluid is an incompressible liquid. An evaluation of the partial derivatives in association with the physical parameters of the system is given in the Appendix.

It can be shown in evaluating the magnitude of each of the S_i that S is always greater than zero. Furthermore, in order to have stable control using feedback one must have S_2 greater than zero. This corresponds to having positive displacement of the storage volume with increasing pressure. The parameter S_1 may be made less than, equal to, or greater than zero depending on the bellows configuration employed. Figures 3-1 and 3-2 illustrate three different configurations corresponding to the three possibilities for S_1 , along with having $S_2 > 0$. The optimum configuration will depend on what variables (heat load, sink temperature, etc.) having the most marked effect on the source temperature. However, as indicated by Equation (2-1), the magnitude of the feedback parameter S_2 should be as large as is practical. In particular S_2 should be large, independently of S and S_1 . The use of an auxiliary fluid which is very sensitive to temperature changes allows one to accomplish this. Both saturated vapors and incompressible liquids qualify. However, the pressure change associated with a given temperature change is several orders of magnitude greater for a liquid than that experienced by a saturated vapor. This is reflected in the relative magnitude of β/χ and $(p/\psi)_a$ for the incompressible liquid and the saturated vapor respectively; and, consequently, results in larger values of S_2 relative to S and S_1 when the auxiliary fluid is an incompressible liquid.

B. Design

A passive feedback controlled heat pipe system has been designed in detail based

on the above considerations and the performance requirements established for a potential Advanced Thermal Control Flight Experiment (ATFE) which is discussed in a later section. The detailed design is shown in Figures 3-3 and 3-4. Design features and performance requirements for this system are summarized in Table 3-1. Basically the design consists of the following:

- (a) Feedback controlled variable conductance heat pipe
- (b) Honeycomb OSR radiator panel
- (c) Simulated equipment shelf

Aluminum saddles are used in mounting the heat pipe to the radiator panel or within the shelf, in order to minimize the thermal resistances.

The radiator and equipment shelf are standard spacecraft items, and therefore only the feedback controlled conductance heat pipe will be discussed at length. The heat pipe consists of a 3/8" O.D. x 0.020" aluminum tube which has an arterial wick inserted. It should be noted that although the pipe is not straight its entire length lies in the same plane in order to permit gravity testing. Methanol was selected as a working fluid because it has a low vapor pressure at the operating temperature and because of its low boiling point. Preliminary tests conducted by Dynatherm have indicated that the sharpest vapor-gas interfaces occur for condensibles at a low vapor pressure. Thus, the use of methanol is intended to reduce diffusion effects. In addition, its low boiling point will prevent freezing in the inactive part of the condenser section except at very low sink conditions. Therefore, the possibility of start-up problems associated with freezing in the condenser section will be reduced. While methanol is

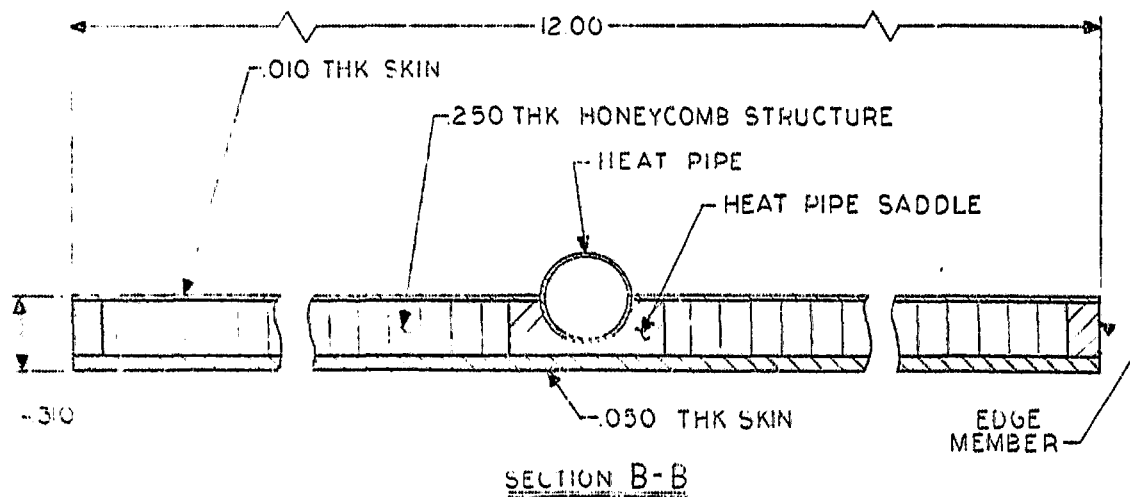
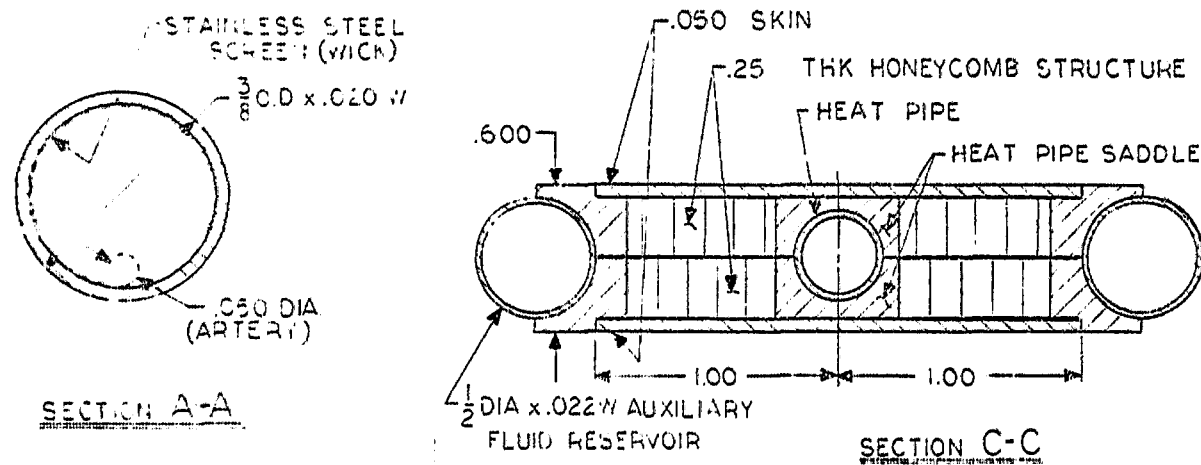


FIGURE 3-4
PASSIVE FEEDBACK CONTROLLED VARIABLE
CONDUCTANCE HEAT PIPE SYSTEM

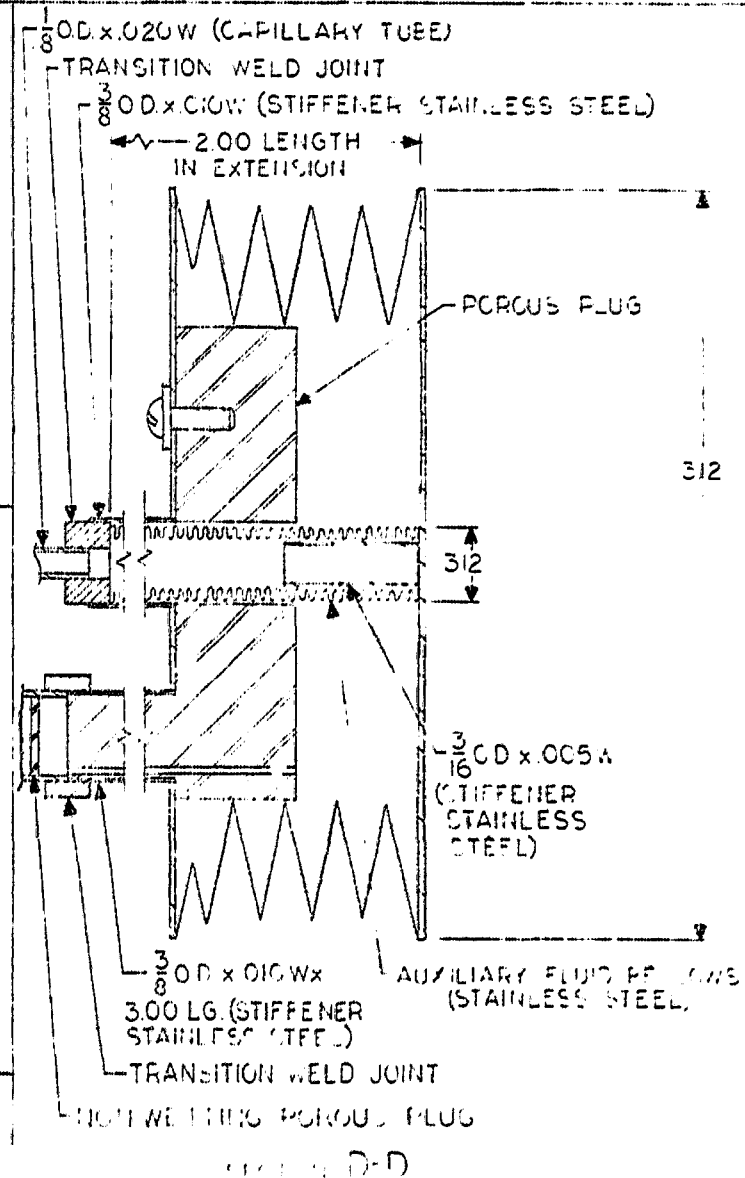


TABLE 3-1
DESIGN FEATURES OF PASSIVE FEEDBACK
CONTROLLED HEAT PIPE FOR ATFE APPLICATION

Heat Pipe Geometry	3/8" O.D. x 0.020" Wall
Evaporator Length	18 inches
Transport Section	6.5 inches
Condenser Length	17 inches
Working Fluid	Methanol

Bellows System

<u>Bellows Parameters</u>	<u>Auxiliary Bellows</u>	<u>Gas Bellows</u>
O.D. (inches)	0.312	3.125
Effective Area (in ²)	0.05	4.89
Stroke (inches)	0.75	0.75
Free Length (inches)	1.65	0.35
Spring Rate (lb/in)	6.0	19.0
Auxiliary Fluid	Liquid Methanol	
Auxiliary Fluid Storage Volume (in ³)	4.0	

Performance Requirements

Nominal Source Temperature (T_s)	65° F
Nominal Sink Temperature (T_o)	360° R
Power Range (Q)	0-30 Watts
Nominal Heat Source Resistance (R_s)	0.5° F/Watt

considered to be a reasonably good working fluid in the low temperature regime, its transport factor is too low to meet the ATFE system's transport requirement with a conventional wick design. The .050" artery, shown in Figure 3-4, which is made from 200 mesh screen has a predicted transport capability in excess of 1500 Watt-inches. This is more than a factor of two (2) greater than that required in the ATFE application. The artery is offset from a 200 mesh screen which is formed around the inside of the tube to provide circumferential distribution of the liquid. Offsetting the artery prevents nucleate bubble formation in the artery and the associated loss of pumping.

The feedback control mechanism and variable storage volume consists of a bellows system whose arrangement is such that S_1 is positive. This was done to provide increased attenuation of any diffusion effects or effects due to changes in the gas temperature that might arise. Standard off-the-shelf bellows were used in the design. The larger outer bellows represents the variable gas storage volume. Centrally located within this bellows is a small auxiliary bellows which contains the incompressible liquid. The auxiliary bellows is connected by a small capillary-like tube to two cylinders (2 cubic inches each), located along the sides of the equipment shelf, which contain the bulk of the auxiliary fluid. Variations in the source temperature (in this case the temperature of the equipment shelf) result in changes in the temperature of the auxiliary liquid thereby causing it to expand or contract and drive the auxiliary bellows. Any displacement of the auxiliary bellows causes an identical linear displacement of the gas bellows, and a variable conductance results. Methanol was chosen as the auxiliary liquid because it has a high ratio of volumetric expansion to isothermal compressibility (β / χ).

As indicated in Figure 3-4 a lightweight hollow plug is located at the front of the gas bellows. Precompressed bellows are used so that their full stroke is realized in extension. Therefore, at the low power/low sink condition the bellows system will experience its minimum displacement. The size of the plug is determined so that it occupies all of the void (except, of course, for that within the bellows convolutions) at this condition. In this way, the bellows expansion required for good temperature control at higher heat loads is reduced and an optimum design is approached. A perforated non-wetting plug is placed at the entrance of the storage volume. Its purpose is to prevent liquid from entering the reservoir while permitting gas to pass through.⁵

There is also a small rod inserted in the auxiliary bellows as well as a sleeve around that portion of the auxiliary bellows outside of the gas bellows. Both of these items are required to eliminate any buckling of the bellows that would occur due to the high internal pressures associated with the incompressible liquid.

C. Steady State Analysis

A parametric analysis was carried out to determine the steady state performance of this system. A sample calculation illustrating the method of analysis is given in the Appendix. The analysis was based on the following assumptions:

- 1.) Radiator area sufficiently large to dissipate the maximum heat load (30 W) to the maximum equivalent sink temperature ($T_o = 360^\circ\text{R}$) at a temperature condition consistent with the design and the maximum load condition. (Nominal heat source temperature $T_s = 65^\circ\text{F}.$)
- 2.) Mass diffusion is negligible.

- 3.) Changes in the temperature of the non-condensable are negligible.
- 4.) Ideal Gas Law applies to the non-condensable.

Assumptions (2.) and (3.) have been introduced to facilitate the analysis. These assumptions can be closely approached in practice through proper design. It will be shown later that the use of feedback results in significant attenuation of variations in the source temperature that would result if these effects were present.

The variation of heat source temperature with changes in the heat input has been determined and is compared to that for an ideal conventional thermal control heat pipe in Figures 3-5 and 3-6. The corresponding variations in the vapor temperature of the working fluid are also shown. The value of feedback control becomes obvious from these Figures. By driving the variable storage volume with a fluid which senses the source temperature the variable volume will be expanded to the extent that the vapor temperature will decrease with increasing heat load. In the conventional thermal control heat pipe dT_v is greater than or equal to zero. Thus since

$$T_s = R_s Q + T_v \quad (2-2)$$

where R_s is the thermal impedance ($^{\circ}\text{F}/\text{W}$) between the heat pipe and heat source, the source temperature will vary directly as the heat load in the conventional system. However, since dT_v can be made negative by using feedback, the variation of source temperature with changes in heat load can be made to go to zero. The comparison indicates a decrease in the variation of the source temperature of approximately 50% in the feedback system for each of the heat source thermal impedances considered.

The effect of variations in the sink temperature on the source temperature was also determined for given heat load and heat source thermal impedance. These results are shown in Table 3-2. As can be seen, the source temperature is essentially insensitive

Figure 3-5

Comparison of Performance Between Passive
Feedback Controlled Heat Pipe & Ideal Conventional
Variable Conductance Heat Pipe

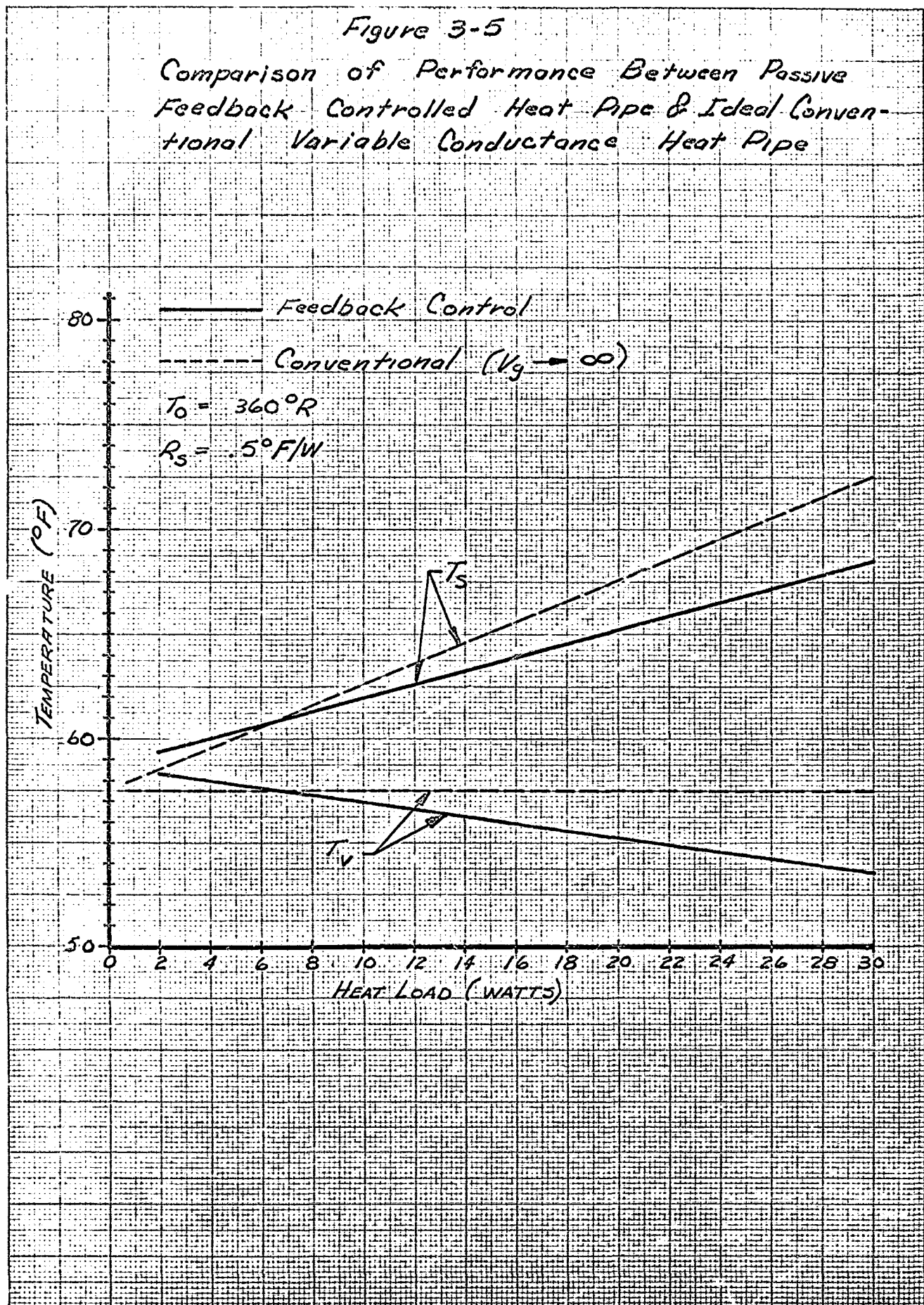


FIGURE 3-6

Comparison of Performance Between Passive Feedback Controlled Heat Pipe & Ideal Conventional Variable Conductance Heat Pipe

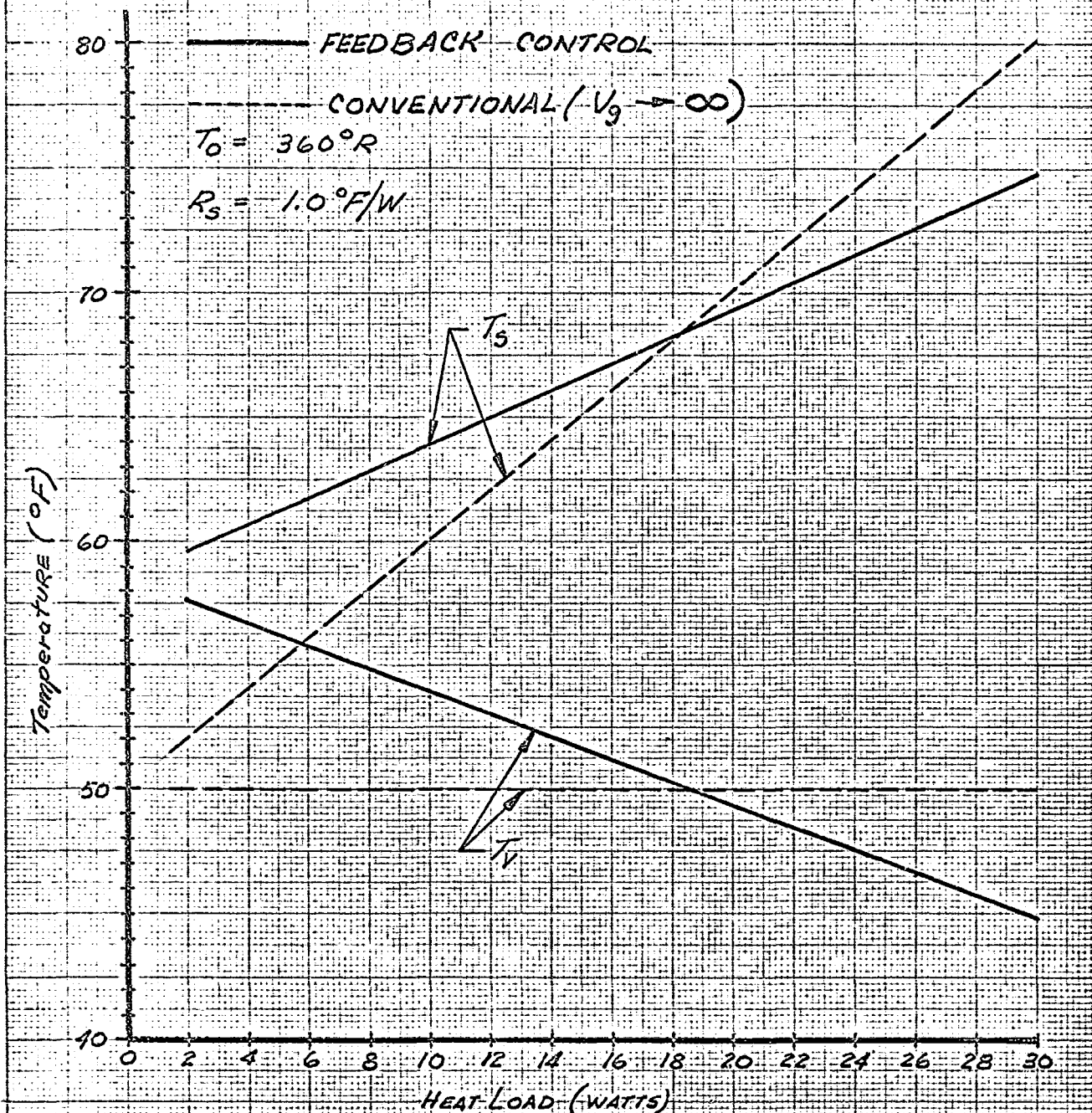


TABLE 3-2

EFFECT OF SINK TEMPERATURE
ON HEAT SOURCE TEMPERATURE
($R_s = 0.5^\circ \text{F/W}$)

Heat Load (Watts)	Heat Source Temperature ($^\circ\text{F}$)	
	Sink Temperature = 360°R	Sink Temperature = 180°R
2	59.4	59.3
6	60.6	60.4
10	61.9	61.6
14	63.2	62.8
18	64.6	63.9
22	65.9	65.2
26	67.3	66.4
30	68.6	67.6

to variations in sink temperature. This would also be the case for an ideal conventional system. An evaluation of the sink temperature term shows that this result is true in general. However, it should be pointed out that this analysis treats mass diffusion and/or changes in gas temperature as independent variables. In reality, these effects could be strongly related to the sink temperature. As an example in a design where a cold storage volume is used (i.e., the storage volume views essentially the effective sink temperature), the temperature of the non-condensable will be essentially identical to the sink temperature. Thus for this case

$$dT_g = dT_o \quad (3-4)$$

and the effect on source temperature as determined from Eq. (3-1) is

$$\frac{dT_s}{dT_o} = \frac{\frac{\partial T_v}{\partial T_o} + \frac{S}{\Psi_v T_o}}{1 + S + S_1 + S_2} \quad (3-5)$$

For radiation heat transfer the first term in Eq. (3-5) becomes

$$\frac{\partial T_v}{\partial T_o} = \left(\frac{T_o}{T_v} \right)^3 \quad (3-6)$$

This term represents the effect of changes in sink temperature on the heat rejection capability and, as mentioned above, will have very little effect on the thermal control characteristics particularly when attenuated by the S_i parameters. However, the second term which relates to the effect of sink temperature on the gas temperature is inversely proportional to the sink temperature. Consequently, since Ψ_v is small ($\sim 10^{-2}/^\circ\text{R}$), the variation of the source temperature with changes in sink temperature could become rather substantial at low sink temperatures unless adequate attenuation is provided by S_1 and S_2 .

In the general case, the attenuation of the variations of source temperature caused by variations in the non-condensable gas temperature or mass diffusion are given by

$$T_g \frac{dT_s}{dT_g} = m_v \frac{dT_s}{dm_v} = \frac{(1+S) \Psi_v}{1+S+S_1+S_2} \quad (3-7)$$

For the ideal conventional thermal control heat pipe, the right side of equation (3-7) becomes equal to Ψ_v . The ratio of the attenuation afforded by feedback to that of an ideal conventional heat pipe ($\frac{1+S+S_1+S_2}{1+S}$) is shown in Figure 3-7 as a function of the heat load for two values of T_o and R_s . These results indicate an increase in attenuation by a factor of two or more for this feedback system.

It should be realized that this particular bellows system was not optimized. Efforts were made in the design to derive the best performance (e.g., the hollow plug in the large bellows); however, no attempt was made to determine an analytical optimization. The many different parameters associated with the system along with the interrelation among many of these parameters (e.g. for a given bellows the spring constant is a function of its free length which is in turn dictated by the required displacement) make an analytical evaluation extremely difficult. However, although this particular system was not optimized it does demonstrate the improvement afforded by a feedback system.

D. Transient Analysis

In general, the use of feedback in a control system reduces the effect of disturbances on the system's output. However, associated with every feedback control system is the problem of potential instability. A stability analysis has been performed for a general passive feedback controlled heat pipe system whose configuration

is the same as that shown in Figure 3-1 (i.e., $S_1 > 0$). The mathematical model consists of the equations used to describe the steady state performance with the inclusion of appropriate transient terms. Transient effects that were included (Cf. Appendix) are:

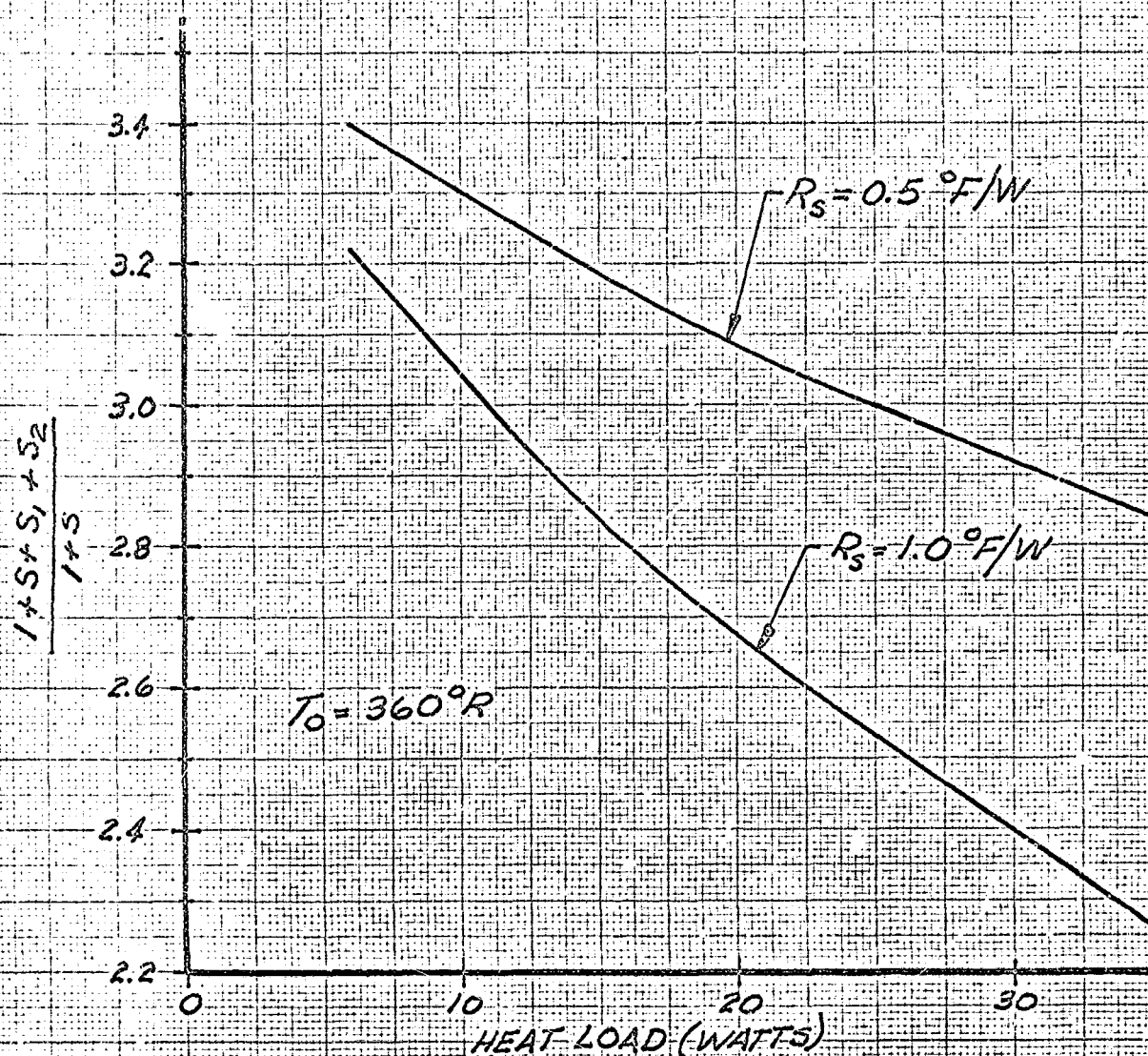
- 1.) Thermal inertia of the heat source ($m c_p$)_s
- 2.) Thermal inertia of the heat pipe system ($m c_p$)_{Hp}
- 3.) Bellows acceleration ($m_B \ddot{\eta}$)
- 4.) Damping of the bellows ($b \dot{\eta}$)

Each of the appropriate equations were linearized and transformed to "Laplace Space." The characteristic equation associated with the response of the source temperature to changes in heat load was then determined. Variations in sink temperature, gas temperature, etc., were not considered in this analysis. The characteristic equation associated with this response is a fourth order polynomial. Depending on the magnitude and sign of the coefficients, the roots of this equation could consist of positive real parts. The existence of such a root results in a transient term which increases exponentially with time and results in an unstable system.

Routh's criteria was used to evaluate the characteristic equation in order to establish what criteria must be satisfied in order to guarantee a stable response. The results indicate that, in general, a passive feedback controlled heat pipe with $S_1 > 0$ will exhibit a stable response to variations in the heat load provided that the viscous damping term is sufficiently large. The magnitude of the damping term required for the ATFE system was determined to be approximately 10^{-3} lb/ft/sec which is extremely small. It appears that, at least for the ATFE application, the damping that is intrinsic to the bellows system would probably be sufficient to guarantee a stable response.

FIGURE 3-7

RATIO OF ATTENUATION AFFORDED BY A
FEEDBACK SYSTEM TO THAT OF AN IDEAL
CONVENTIONAL THERMAL CONTROL HEAT
PIPE VS. HEAT LOAD



Section 4

ACTIVE FEEDBACK CONTROLLED HEAT PIPE SYSTEM

The active feedback controlled heat pipe, like the passive system, uses the principle of variable conductance to provide thermal control. In both systems the variable conductance is effected by varying the effective storage volume available for a non-condensable gas. Instead of a bellows system and an auxiliary fluid reservoir which was used in the passive system, an electrical temperature sensor, an electronic controller, and a heated storage volume are used to provide active control.

A schematic of an active control system is shown in Figure 4-1. The storage volume is located at the condenser end of the heat pipe. A continuous wick extends into the gas storage volume. As a result, saturated working fluid is always present in liquid form in the storage volume. The partial pressure of the vapor in equilibrium with the liquid is determined by the temperature of the storage reservoir. Thus, the partial pressure and therefore the partial volume occupied by the non-condensable in the reservoir is determined by the vapor pressure and ultimately by the temperature of the reservoir. Since the vapor pressure of any fluid is sensitive to temperature, only small variations of the storage temperature suffice to achieve large changes of the effective volume occupied by the non-condensable. These changes result in movement of the gas out of the reservoir into the condenser (or vice versa), thereby changing the conductance of the heat pipe. In an active feedback system, the heat source is monitored electrically and the signal drives a small auxiliary heater located within the storage volume. In this way, the storage temperature and therefore the variable conductance is related to the source temperature and feedback control is obtained.

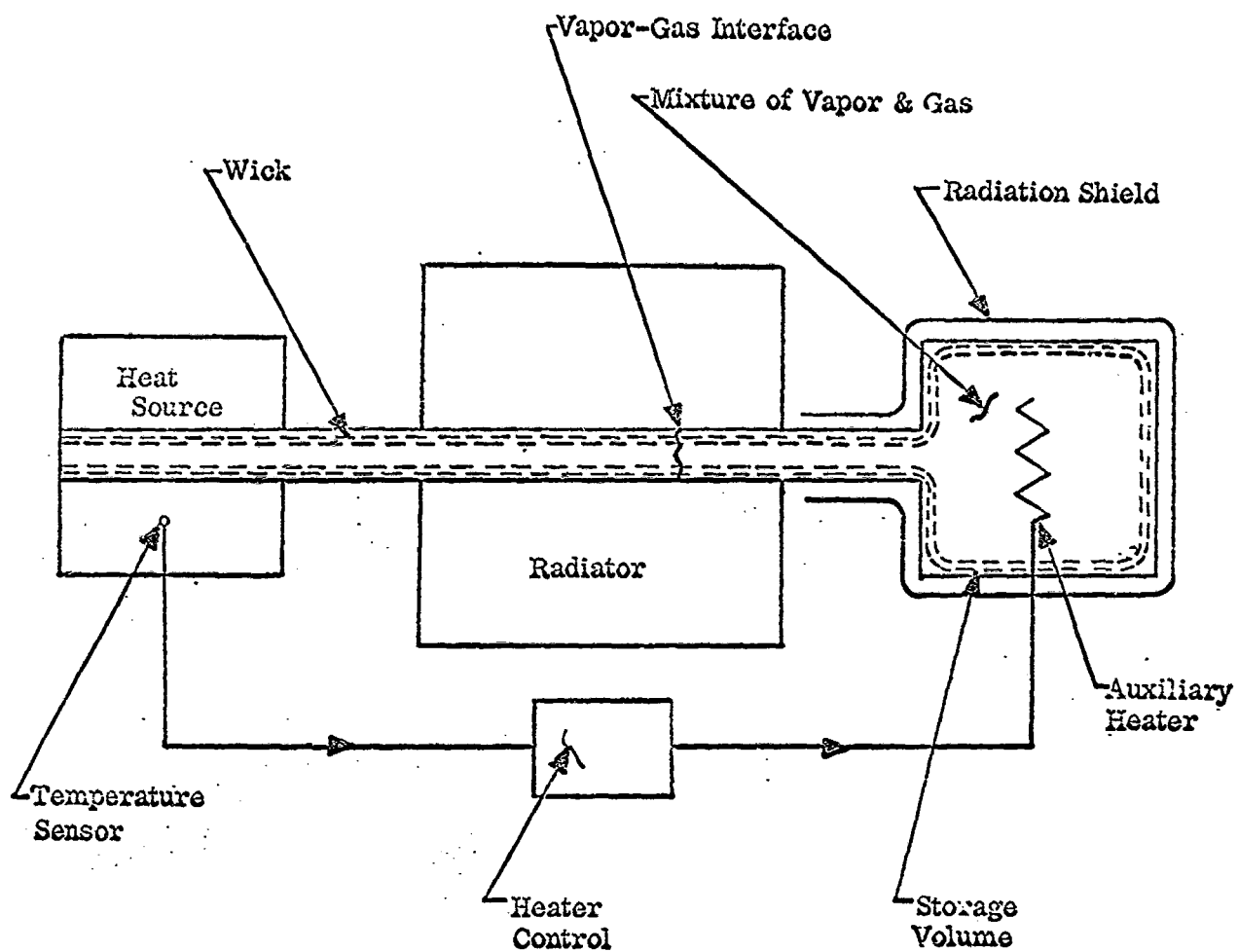


FIGURE 4-1

ACTIVE FEEDBACK CONTROLLED VARIABLE
CONDUCTANCE HEAT PIPE SYSTEM

A. Analysis

A steady-state analysis has been performed to evaluate the control capability along with storage volume requirements for active feedback systems. The following assumptions were made in performing the analysis:

1. The non-condensable gas obeys the ideal-gas equation of state.
2. Mass diffusion is negligible--i.e., a sharp interface exists between the working fluid vapor and the non-condensable gas at the beginning of the inactive part of the condenser.
3. The entire condenser length is active at the high power condition.
4. The inactive part of the condenser is at the sink temperature T_o .
5. Conduction along the heat pipe wall is negligible.

Using these assumptions, the following equations apply:

1. Conservation of Mass

$$m_c + m_{st} = m_g \quad (4-1)$$

2. Law of Additive Pressures (applicable to inactive part of the condenser and also the storage volume)

$$\pi + p_g = p_v \quad (4-2)$$

3. Ideal-Gas Equation of State

$$p_g V = m R_g T \quad (4-3)$$

Thus for the high power condition

$$\left(\frac{p_v - \pi_{st}}{R_g T_{st}} \right)_{H} V_{st} = m_g \quad (4-4)$$

for the low power condition

$$\left(\frac{p_v - \pi_o}{R_g T_o} \right)_L V_c + \left(\frac{p_v - \pi_{st}}{R_g T_{st}} \right)_L V_{st} = \frac{m}{g} \quad (4-5)$$

Combining Eqs. (4-4) and (4-5) and solving gives

$$\frac{V_{st}}{V_c} = \frac{\left(1 - \frac{\pi_o}{p_v} \right)_L \left(\frac{T_{st}}{T_o} \right)_L}{\left(\frac{p_{vH}}{p_{vL}} \right) \left(1 - \frac{\pi_{st}}{p_v} \right)_H \frac{T_L}{T_H} - \left(1 - \frac{\pi_{st}}{p_v} \right)_L} \quad (4-6)$$

Thus assuming that the vapor temperature and the temperature of the storage volume are specified as input for the high and low power condition the above equation can be solved to give V_{st} for a given condenser volume (V_c).

Analysis of Eq. (4-6), along with a recognition of how the variable conductance heat pipe operates, indicates what the storage temperature should be at the high and low power condition.

- (1) At the high power condition all of the non-condensable gas should be contained in the storage volume. This implies that the partial pressure of the working fluid in the storage volume approach zero, which in turn implies that the storage temperature approach the melting temperature of the working fluid. This can be accomplished by selecting a fluid whose freezing point is near the sink temperature.
- (2) At the low power condition ideally, all of the non-condensable should be in the condenser section. This implies that the partial pressure of the working fluid in the storage volume approach the system vapor

pressure, which in turn implies that the storage temperature approach the vapor temperature of the working fluid in the heat pipe. However from a practical consideration, in order to minimize mass diffusion and its degradation on temperature control using variable conductance techniques, the storage temperature should be somewhat lower than the heat pipe vapor temperature. This allows some concentration of the non-condensable to exist within the storage volume thereby reducing the potential for mass diffusion. Acceptable storage temperatures at the low power condition must be determined from the diffusion characteristics for the system which is beyond the scope of this program.

As before, the storage volume requirement is related to the heat load transported by the heat pipe by

$$T_s = T_v + R_s Q \quad (2-2)$$

In typical applications the allowable source temperature variation will be specified for a specified heat load variation. Then the required variation in vapor temperature is known from Eq. (2-2) as a function of the heat load Q . Thus, storage volume requirements can be related to a specified variation in heat load through Eqs. (4-6) and (2-2) along with the equation of state for the condensable working fluid. The above analysis allows the determination of optimum storage volume requirements for a specified variation of source temperature and heat load for a given heat pipe design. With the storage volume known, Eq. (4-4) can be solved to determine the mass of the non-condensable required.

The ratio (V_{st}/V_c) has been determined as a function of $R_s Q_{max}$ for a given

sink temperature ($T_o = 200^\circ\text{K}$). The results are shown for methanol and ammonia in Figures (4-2) and (4-3) respectively. The following parameters were specified as follows in determining these results.

- (1) $T_s = 65^\circ\text{F} = 291^\circ\text{K}$
- (2) $dT_s = 0^\circ\text{K}$
- (3) $T_{v_L} = T_s$ (i.e., $Q_{\min} \approx 0$)
- (4) $T_{st_H} = T_o$

As indicated by the Figures, the results were determined for the ideal storage temperature at the low power condition ($T_{st_L} = T_{v_L}$, i.e., assuming no mass diffusion problem) and a storage temperature 10°F less than the vapor temperature for the low power case. The latter value of storage temperature was selected only to show the effect of reducing this temperature; it is by no means necessarily an optimum storage temperature. The optimum can be determined only by evaluating mass diffusion profiles. The results indicate that for a given T_{st_L} and $R_s Q_{\max}$ the storage volume requirements are greater for methanol than for ammonia. Also decreasing the storage temperature by 10°F for the low power condition results in increased storage volume requirements and a lower maximum allowable $R_s Q_{\max}$ for both working fluids. The storage volume requirements are greater and increase more rapidly with increasing $R_s Q_{\max}$ for methanol than for ammonia because the former fluid has a larger percentage change in vapor pressure for a given change in temperature.

B. Design

A detailed design of an active feedback controlled heat pipe system has been completed. This design is based on the Advanced Thermal Control Flight Experiment

FIGURE 4-2
STORAGE VOLUME REQUIREMENTS FOR
ACTIVE FEEDBACK CONTROL

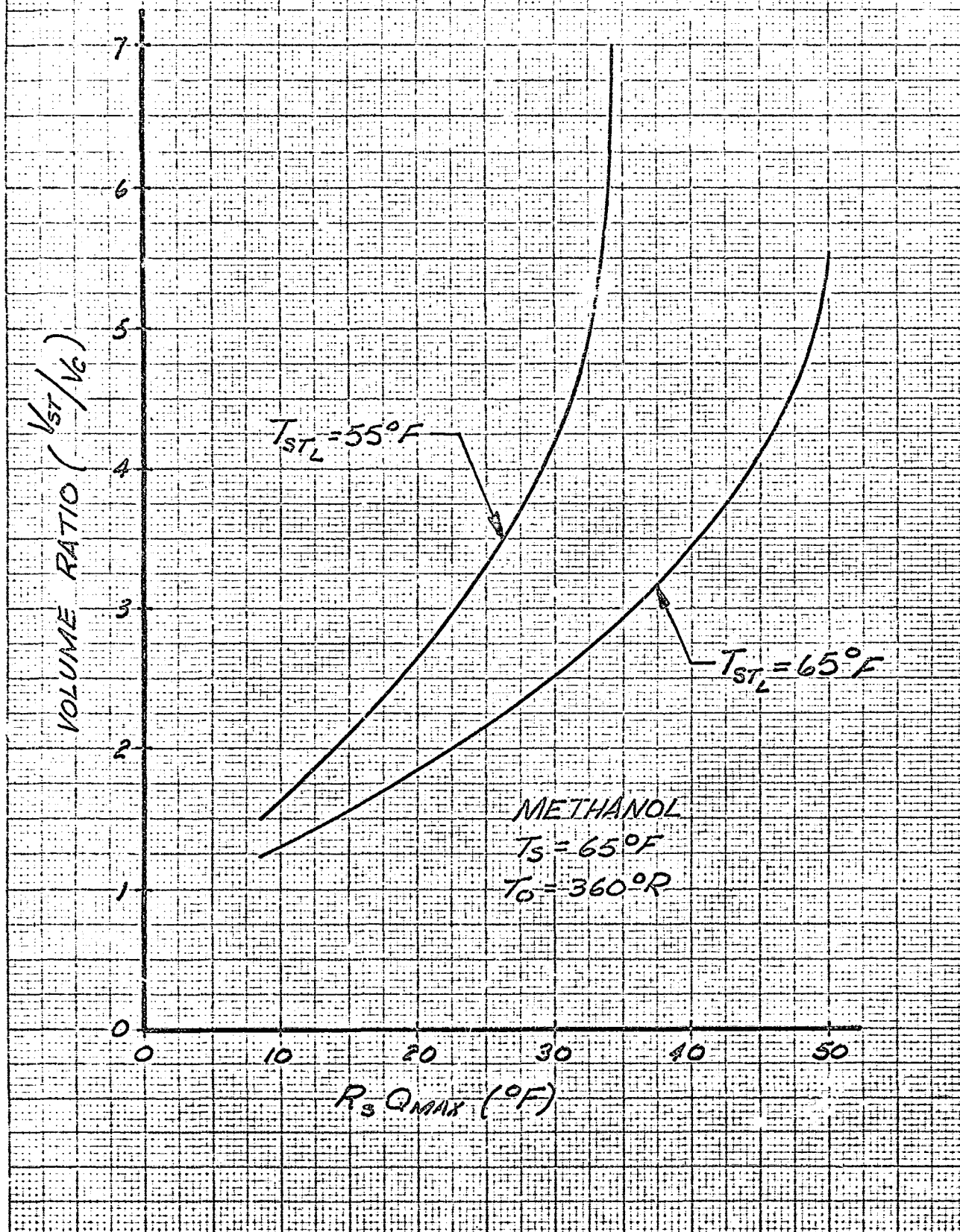
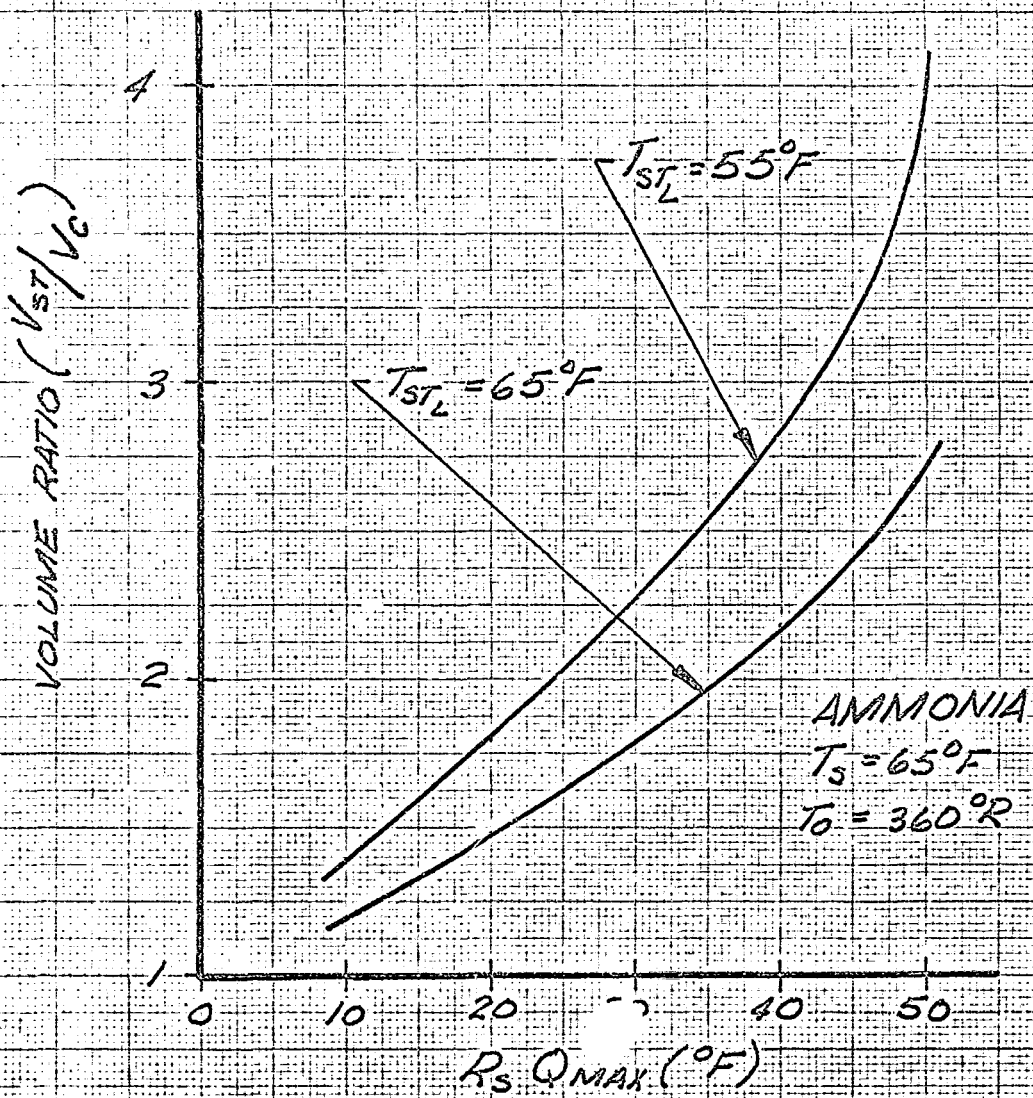


FIGURE 4-3

STORAGE VOLUME REQUIREMENTS FOR
ACTIVE FEEDBACK CONTROL



(Cf. Section 5) and is shown in Figures 4-4 and 4-5. As can be seen, the active system is identical to the passive, except for the feedback control mechanism.

In this case, instead of a bellows, the storage volume is a 5.0 in³ aluminum cylinder which is brazed to the heat pipe at the end of the condenser section. Two layers of 200 mesh screen line the inner wall of the cylinder to distribute the working fluid within the storage cavity. This wick is intermeshed with the peripheral wick in the heat pipe in order to prevent accumulation of the working fluid in either section. A heater is wrapped around the outside of the storage volume to provide temperature control of the saturated fluid and therefore control of the partial volume occupied by the non-condensable. Radiation shields cover the heater in order to reduce the losses to space and therefore the auxiliary heater power requirements. For this particular design and application, the auxiliary power requirement is less than one (1) watt for an emittance of the radiation shield as high as 0.30.

The auxiliary heater is driven by an electronic controller which transforms the input signal from the heat source temperature sensor. All active elements of the controller are solid state and can be expected to operate reliably. The controller circuitry consists of a bridge circuit which is actuated by a thermistor type sensor, a variable gain control amplifier, and a low power SCR firing circuit. Although a controller design has not been finalized for ATFE application, the typical envelope size for this type of controller is 3" x 3" x 1".

Variations in the heater power result in changes in the amount of non-condensable gas in the storage volume. This in turn results in a variable conductance as the gas front moves into or out of the heat pipe. A change in the conductance results in an adjustment of the vapor temperature which in turn adjusts the source temperature, and the feedback loop is completed.

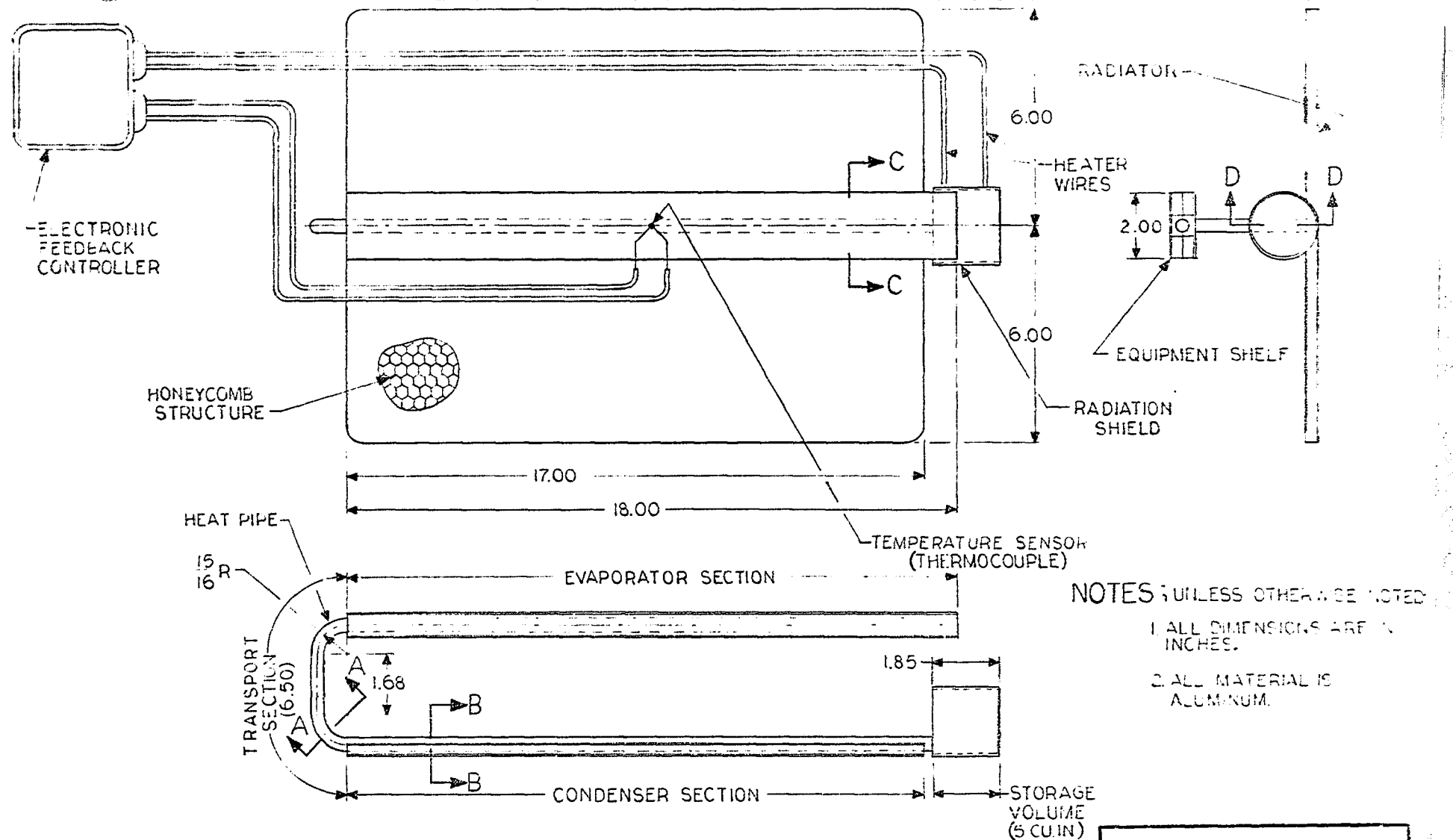
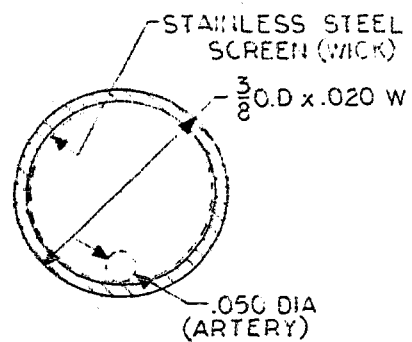
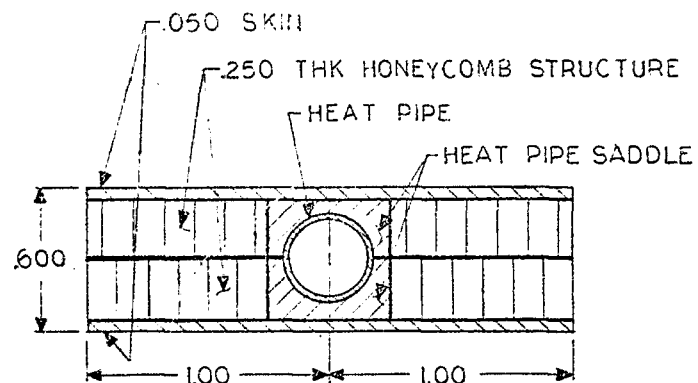


FIGURE 4-4
ACTIVE FEEDBACK CONTROLLED VARIABLE
CONDUCTANCE HEAT PIPE SYSTEM

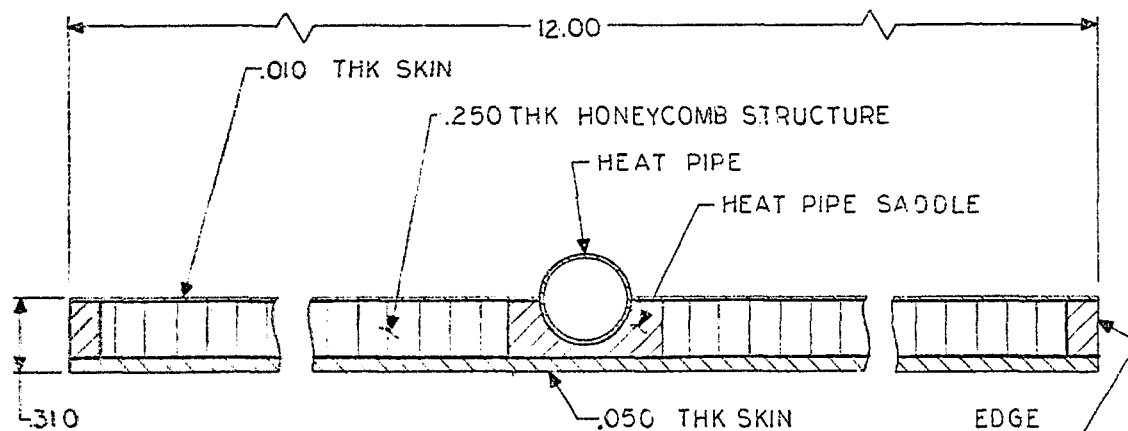
dynatherm CORPORATION



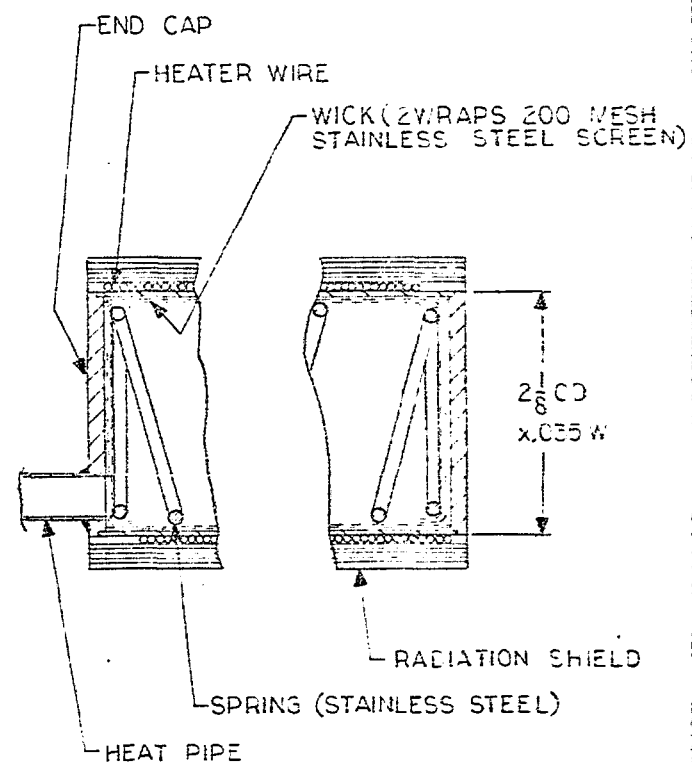
SECTION A-A



SECTION C-C



SECTION B-B



SECTION D-D

FIGURE 4-5
ACTIVE FEEDBACK CONTROLLED VARIABLE
CONDUCTANCE HEAT PIPE SYSTEM

Section 5

ADVANCED THERMAL CONTROL FLIGHT EXPERIMENT

The results of efforts completed under this program demonstrate that both active and passive feedback controlled variable conductance heat pipes are feasible and can provide significant improvement over existing methods of thermal control. Consequently, a potential future application was defined and evaluated. Initially, this application would be considered as an Advanced Thermal Control Flight Experiment (ATFE) in which various thermal control devices including a feedback controlled heat pipe would be evaluated simultaneously. In an actual application this concept could be used to provide temperature control of spacecraft equipment shelves.

In addition to a feedback controlled heat pipe, the proposed thermal control system utilizes fusible material and a thermal diode device. Figure 5-1 presents a schematic of the flight experiment. A conceptual design of the experimental package is shown in Figure 5-2. Solar illumination of the absorber panel supplies the thermal power to the experiment. Solar energy is used in order to minimize power demands by the experiment on the spacecraft. The experiment is assumed to be mounted on a synchronous orbiting spacecraft. The corresponding solar flux profile is shown in Figure 5-3. Heat load and sink conditions associated with this orbit were determined to be more than adequate for evaluating the thermal control capability of the various components.

Inside the package, a thermal diode and feedback controlled heat pipe couple the simulated equipment shelf to the absorber and radiator panels respectively. During the period of occultation, the freezing of the fusible material keeps the shelf temperature within the desired operating range and also provides enough heat to maintain a vapor-gas interface within the variable conductance heat pipe. The individual T/C components were

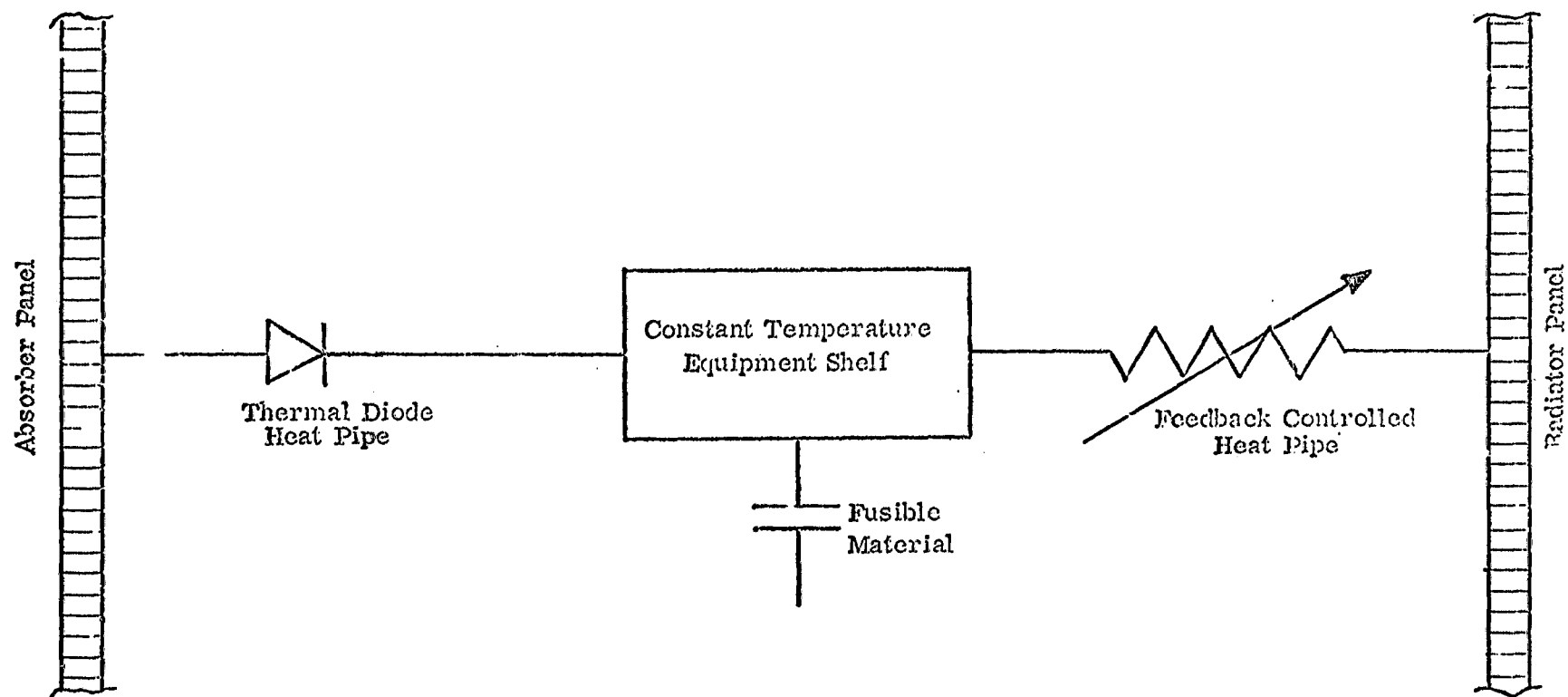


FIGURE 5-1
SCHEMATIC OF ADVANCED THERMAL CONTROL EXPERIMENT

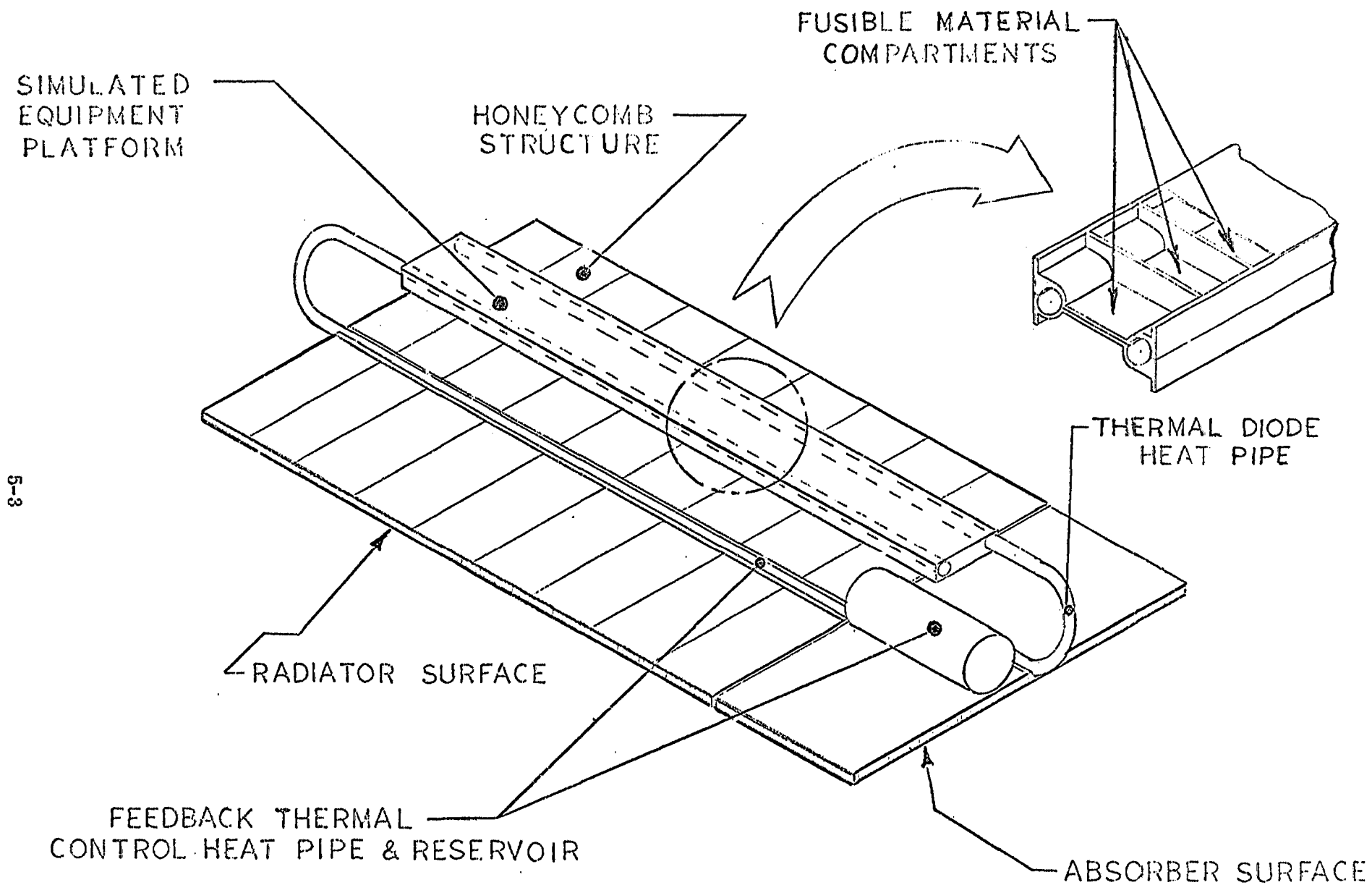
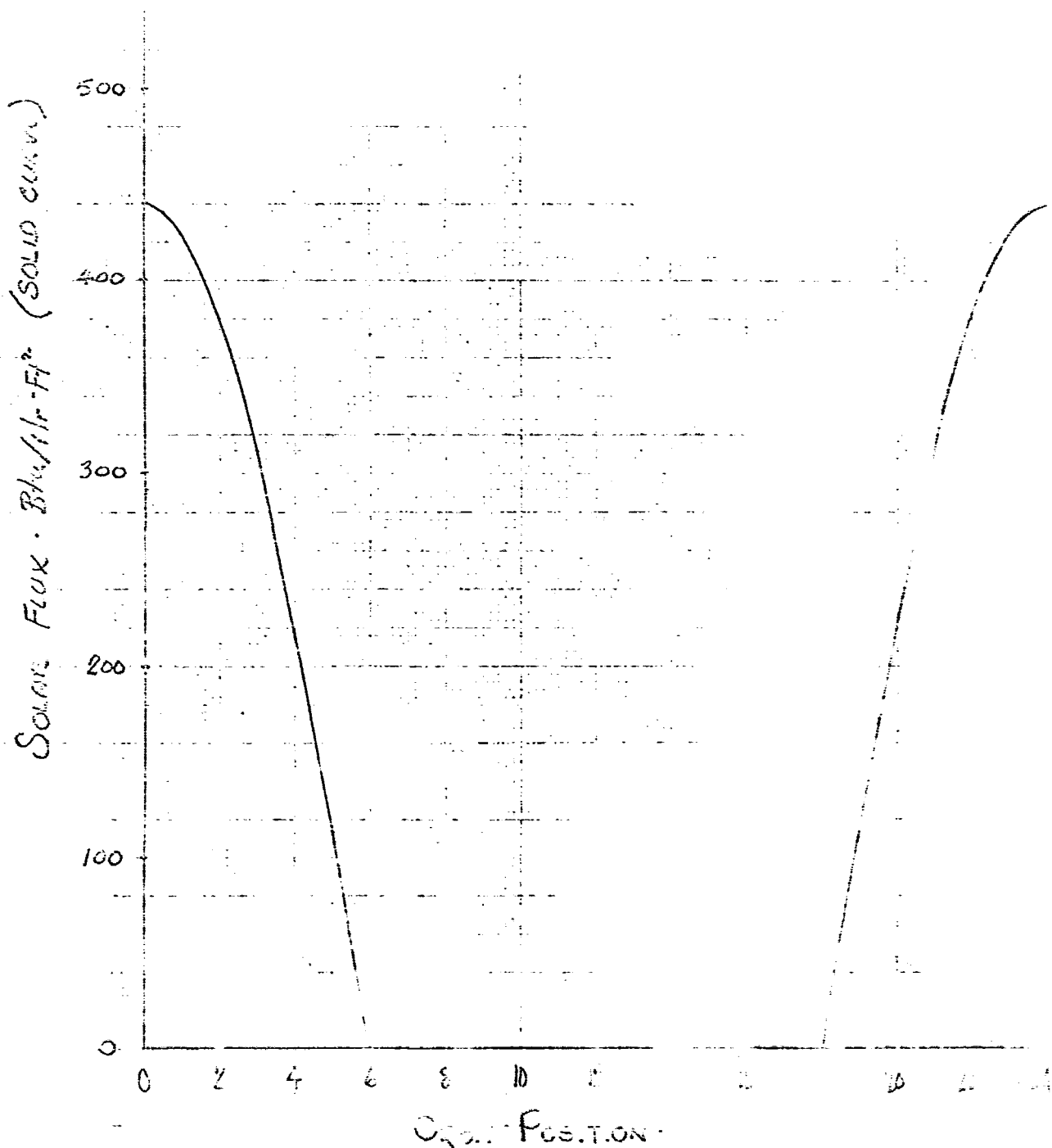


FIGURE 5-2
ADVANCED THERMAL CONTROL TECHNOLOGY EXPM'T

FIGURE 5-2

Solar Illumination Heat Transfer



designed to maintain the equipment shelf at 65°F at all times. A super insulation blanket isolates the experiment from the spacecraft thermally. Only instrumentation penetrates the experiment envelope.

A. Description of ATFE System

The proposed experiment package was designed to conform to an envelope size of 24" x 12" x 5". There are seven major components which make up this system.

These are: (Cf. Figure 5-2)

- (1) Absorber Panel
- (2) Radiator Panel
- (3) Thermal Diode Heat Pipe
- (4) Feedback Controlled Variable Conductance Heat Pipe
- (5) Fusible Material
- (6) Simulated Equipment Platform
- (7) Super-Insulation Blankets.

A discussion of these components follows.

1. Absorber and Radiator Panels

The absorber and radiator panels are mounted adjacent to each other and in the same plane. Both of these panels are fabricated from 1/4" aluminum honeycomb structure. For maximum solar absorption, the thermal coating of the absorber system must have:

- (a) A high solar absorptance, and
- (b) A high ratio of solar absorptance to hemispherical emittance.

In contrast, the radiator system should have high emittance and a low ratio of $\frac{\alpha}{\epsilon}$ for maximum heat rejection. An analysis was performed which evaluated

different thermal coatings in order to optimize the space viewing side (24" x 12") to provide maximum heat absorption to the experiment at the high noon orbital position. The following thermal coatings were considered in the analysis:⁷

(1) Inconel X foil	$\alpha/\epsilon = 0.66/0.10$
(2) Cat-A-Lac Black	$\alpha/\epsilon = 0.96/0.85$
(3) Honeywell Ce O ₂ /Mo/Mg F ₂	$\alpha/\epsilon = 0.85/0.11$
(4) ZnO in Potassium Silicate (Z-93)	$\alpha/\epsilon = 0.20/0.90$
(5) Optical Solar Reflector Mirrors (OSR)	$\alpha/\epsilon = 0.05/0.80$

Results of this analysis indicate that a maximum heat input/dissipation of 28 Watts can be obtained using absorber and radiator panels coated with the Honeywell Ce O₂/Mo/Mg F₂ and OSR's respectively. Both thermal coatings have exhibited stable behavior under simulated and actual flight conditions^{7, 8} and are considered acceptable for this experiment. The respective optimum areas were determined to be 40 and 204 square inches.

2. Thermal Diode Heat Pipe

The thermal diode heat pipe is a device that will transfer heat with a high thermal conductance in only one direction. This device will be used to transfer heat from the absorber to the equipment shelf. Its evaporator section will be attached to the absorber panel ($L_e = 3.34''$) while the condenser is embedded in the simulated equipment shelf ($L_c = 18''$). During the shadow portion of the orbit, the absorber panel will emit thermal energy causing the temperature potential to reverse directions. At this point, in order to avoid heat loss from the equipment platform and because of the nature of its operation, the diode ceases to transfer heat. Further details of the design and construction of this

thermal diode are beyond the scope of this contract.

3. Feedback Controlled Variable Conductance Heat Pipe

During the illuminated portion of the orbit, the solar energy absorbed by the experiment varies according to the sine of the angle between the panels and the rays of the sun. In order to maintain the equipment shelf at constant temperature throughout this period, a feedback controlled variable conductance heat pipe couples the shelf to the radiator panel. The detailed designs and performance of an active and a passive feedback controlled heat pipe suitable for this application have been discussed in the previous sections. In essence, the feedback control varies the heat rejection area to maintain a balance between the absorbed and emitted thermal energy at the desired operating temperature. The evaporator region of this heat pipe is embedded in the equipment platform ($L_e = 18''$) while the condenser portion (which varies in length between zero and 17 inches) is attached along the length of the segmented radiator panel. Extending from the end of the condenser is a storage volume containing the non-condensable gas employed to control the heat rejection area of the heat pipe.

4. Fusible Material

A fusible material is used to compensate for energy losses from the equipment shelf during shadow periods. In addition, a minimum amount of energy from the fusible material is also utilized to sustain a vapor-gas interface within the variable conductance heat pipe during these periods. This is necessary in order to guarantee rapid start-up of the heat pipe system.

During the initial solar energy absorption stage, the heat melts the fusible material. This energy is stored until it is required for thermal control

during the occult period. Upon freezing, energy is released to compensate for the previously mentioned losses.

The fusible material selected for this application is one of the normal paraffins, hexadecane ($C_{16}H_{34}$). Reasons for selecting this type of material are as follows:

- (a) It has a high heat of fusion (102 Btu/Lbs).
- (b) It is a chemically inert and stable compound.
- (c) It is nontoxic and noncorrosive.
- (d) Its physical properties, including low vapor pressure and volume reduction during solidification, are conducive to package design and zero-g operation.
- (e) It has a well defined melt temperature ($T_m = 64^{\circ}F$) which is suitable for the thermal control of many spacecraft components.

For the proposed application, seven tenths of a pound of hexadecane packaged inside the simulated equipment shelf will be required.

5. Simulated Equipment Shelf

The simulated equipment shelf (or platform) is shown in Figure 5-4. It has an envelope of 18" in length, 2" in width, and 1" in depth. Two heat pipes (diode and feedback) are embedded inside the platform. The platform is fabricated from a cast aluminum webbed structure. The webbed structure is used to reduce the thermal resistance between the heat pipe, and at the same time, provide containment of the fusible material. The individual webs provide a large surface area in contact with the fusible material thereby promoting complete melting/freezing. In addition to the inherent stiffness of the heat pipes,

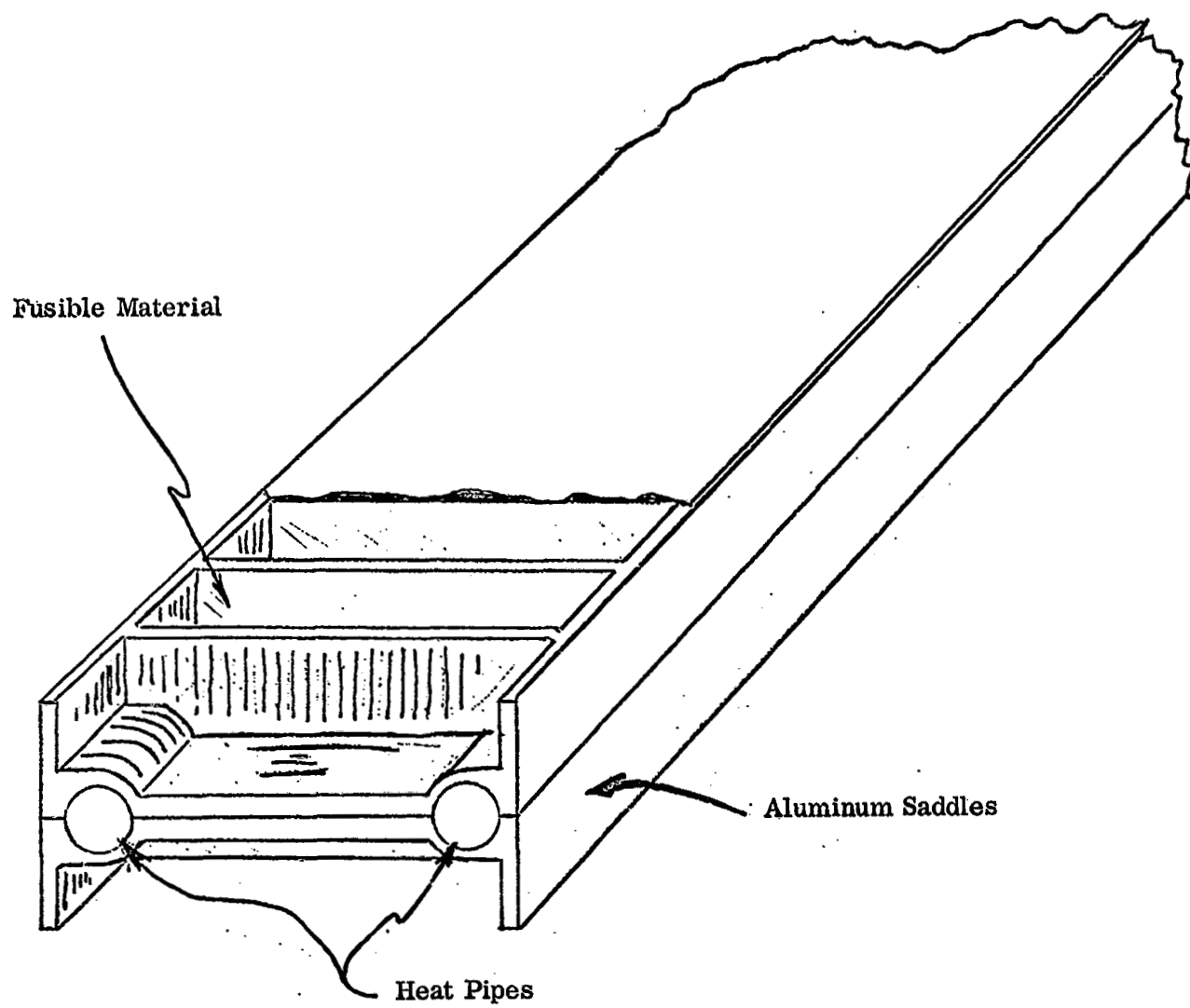


FIGURE 5-4
SIMULATED EQUIPMENT SHELF

a support structure will be provided between the shelf and absorber/radiator panels.

6. Super-Insulation Blankets

Super-insulation blankets will be attached to the support structure and panel edges to eliminate thermal interchange between the experiment and the spacecraft. When the experiment is completely assembled, the only surfaces visible other than insulation will be the thermal control coatings. The blankets will be fabricated from 25 to 30 layers of aluminized mylar super-insulation.

A summary of the design features of the various components is presented in Table 5-1. The total weight of the system has been calculated to be less than six pounds. A weight breakdown is presented in Table 5-2.. The weight for the feedback controller was estimated to be approximately the same for either an active or passive system.

B. Performance of ATFE System

1. Steady State Analysis

A steady state analysis was performed to determine the performance of the ATFE system. The following factors are essential in establishing the optimum radiator/absorber system for maximum heat input:

- (1) The magnitude of the thermal conductance between the equipment shelf and absorber/radiator panels.
- (2) The effect of aluminum substrate width and thickness on the fin efficiencies of the absorber/radiator panels.

At the high noon position during a synchronous orbit, the steady state heat balance equations for the absorber and/or radiator panels are:

TABLE 5-1
DESIGN SUMMARY OF ATFE SYSTEM

Experiment Envelope	24" x 12" x 5"
Absorber	
1/4" Aluminum Honeycomb Panel	3.34" x 12"
Thermal Coating - Ce O ₂ /Mo/Mg F ₂	$\frac{\alpha}{\epsilon} = \frac{0.85}{0.11}$
Radiator	
1/4" Aluminum Honeycomb Panel	17" x 12"
Thermal Coating - OSR	$\frac{\alpha}{\epsilon} = \frac{0.05}{0.80}$
Thermal Diode	
3/8" O.D. Aluminum Tube	
Evaporator Length	3.34"
Transport Length	6.5"
Condenser Length	18"
Fusible Material	Hexadecane
Feedback Controlled Heat Pipe	
3/8" O.D. Aluminum Tube	
Evaporator Length	18"
Transport Length	6.5"
Condenser Length	17"
Equipment Platform - Cast Aluminum	18" x 2" Cast
Insulation	Aluminized Mylar Super-Insulation

TABLE 5-2
WEIGHT SUMMARY OF ATFE SYSTEM

	<u>Item</u>	<u>Weight (Lbs.)</u>
1.	Absorber/Radiator Panels	2.17
2.	Feedback Controlled Heat Pipe and Aluminum Saddles	0.50
3.	Feedback Controller (Passive or Active)	0.25
4.	Thermal Diode and Aluminum Saddles	0.28
5.	Fusible Material	0.70
6.	Equipment Platform	1.04
7.	Mounting and Support Structure	0.50
8.	Insulation Blankets	<u>0.36</u>
	Total	5.80

$$\left(\alpha_p G - \epsilon_p \sigma T_p^4 \right) A_p - K (T_p - T_S) = 0 \quad (5-1)$$

$$Q_{\text{net}} = K (T_p - T_S) \quad (5-2)$$

where $Q_{\text{net}} > 0$, net heat transfer absorbed (by absorber)

$Q_{\text{net}} < 0$, net heat transfer rejected (by radiator)

$\alpha_p, \epsilon_p, A_p, T_p$ = Solar absorptance, emittance, area to space, and temperature of panel (absorber or radiator)

K = Effective thermal conductance between equipment shelf and panel

T_S = Temperature of equipment shelf (65°F)

G = Solar flux constant (440 Btu/hr-ft²)

These equations were evaluated parametrically to determine the effect of thermal conductance on net heat transfer. The results are presented in Figures 5-5 and 5-6 for the absorber and radiator panels, respectively.

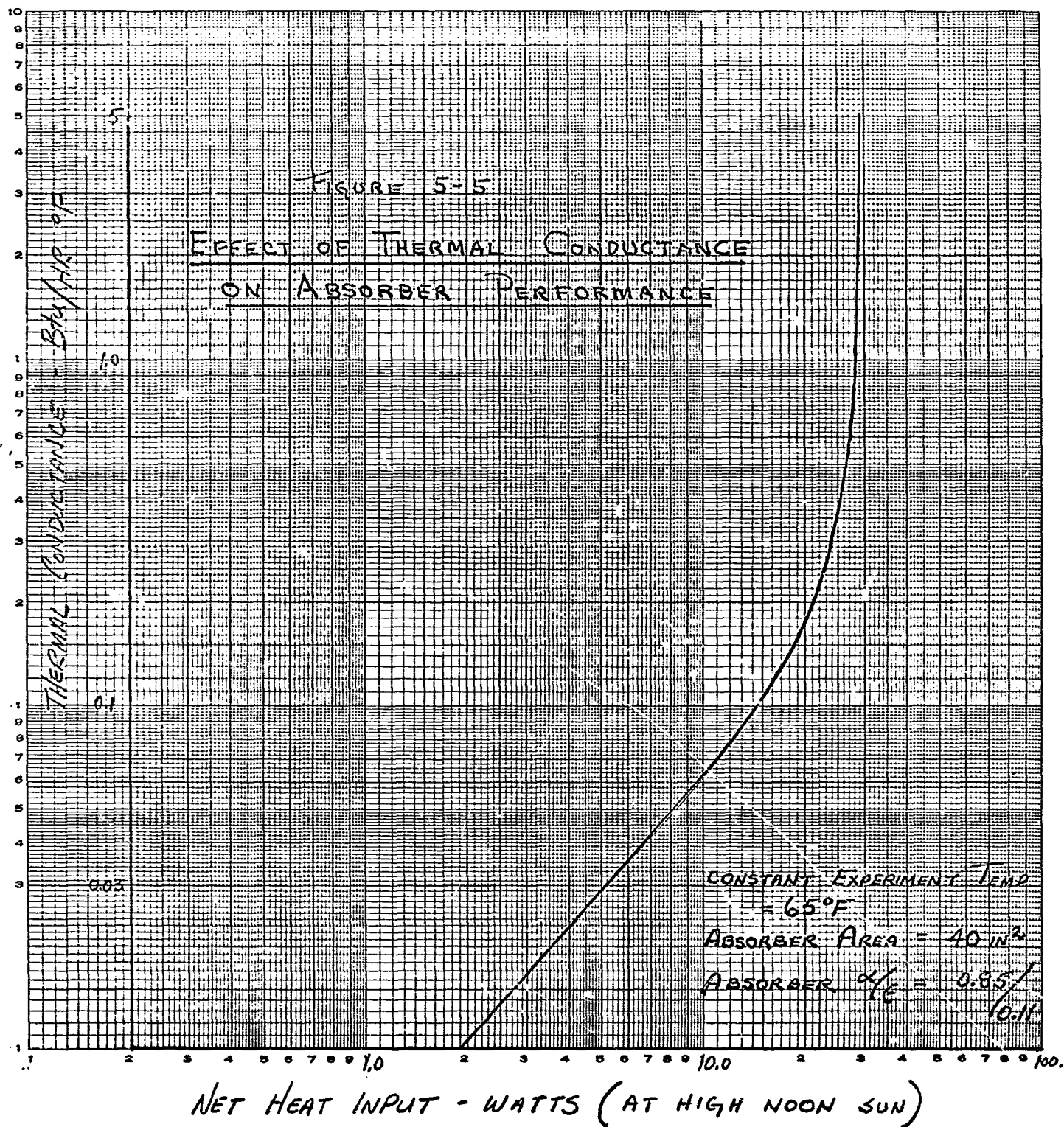
For the ATFE configuration under evaluation, the following results are apparent.

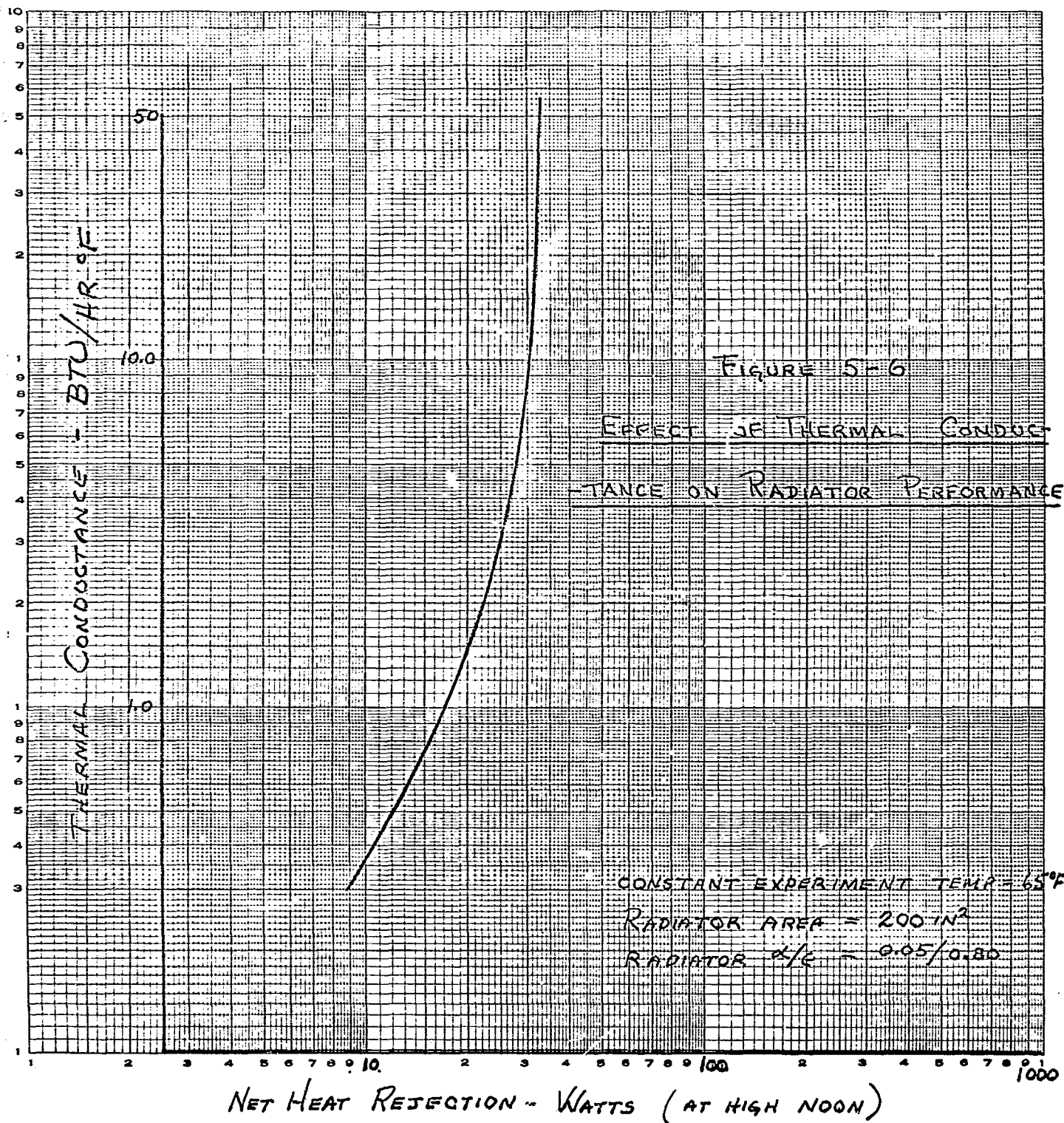
- (a) Maximum capabilities for both the absorber and radiator panels are achieved if the conductance between them and the equipment shelf are greater than 1.0 and 10.0 Btu/Hr-°F, respectively.
- (b) The hypothetical maximum capabilities (infinite conductance to shelf) at the high noon position-synchronous orbit are:

net heat absorption = 29.3 watts and

net heat rejection = 33.3 watts.

An illustration of the model used to determine the effect of the absorber and radiator panels substrate thicknesses is shown in Figure 5-7. The temperature variation across the width of the panel is given by (Ref. 5-1)





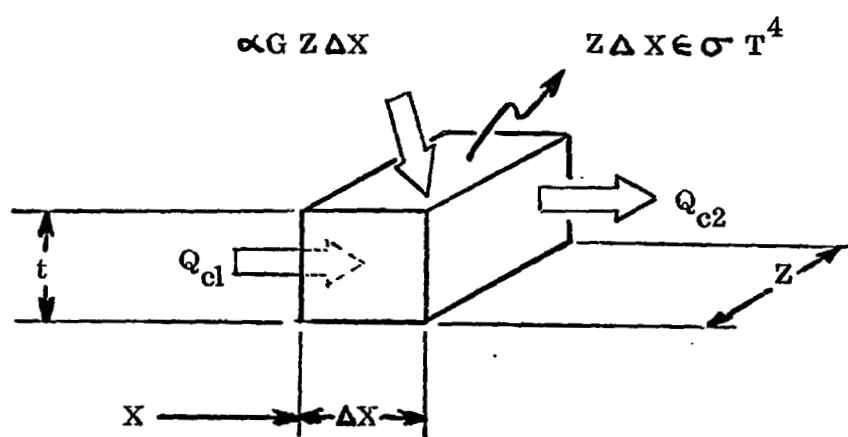
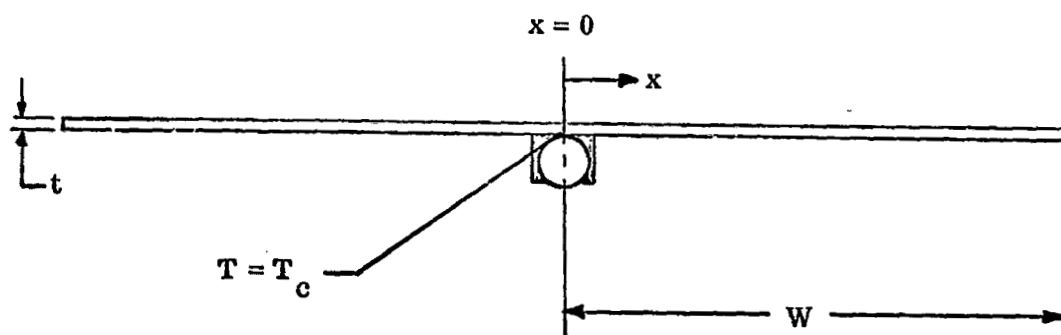


FIGURE 5-7
THERMAL MODEL FOR FIN EFFICIENCY

$$T = T_c \left(.75 + a/b + (.25 - a/b) (\cosh \sqrt{b} X - \tanh W \sqrt{b} \sinh X \sqrt{b}) \right) \quad (5-3)$$

and the fin efficiency is

$$\eta_f = \frac{T_c t \sqrt{b} (.25 - a/b) \tanh W \sqrt{b}}{(\epsilon \sigma T_c^4 - \alpha G) W} \quad (5-4)$$

where:

$$a = \frac{\alpha G}{k t T_c}$$

$$b = \frac{4 \sigma \epsilon T_c^3}{k t}$$

T = Temperature at Location X

T_c = Temperature at the heat pipe condenser section

W = Distance from heat pipe to edge of panel

k = Thermal conductivity of skin

t = Skin thickness

The fin efficiency and maximum temperature gradient are shown in

Figure 5-8 as a function of aluminum substrate thickness. Because of the high solar absorptance and low emittance of the absorber panel, its efficiency is not affected by temperature gradient. As a result the absorber fin efficiency is calculated to be over 95% for any practical thickness. Based on structural considerations the absorber thickness will be 0.04" ($\eta_f = 97\%$). Unlike the absorber, the radiator heat rejection capability is a function of the panel temperature distribution. Its efficiency increases with a large slope with increasing thickness until a value of 0.09" ($\eta_f = 92\%$) is reached. At this point increased thickness has a minimum effect on efficiency.

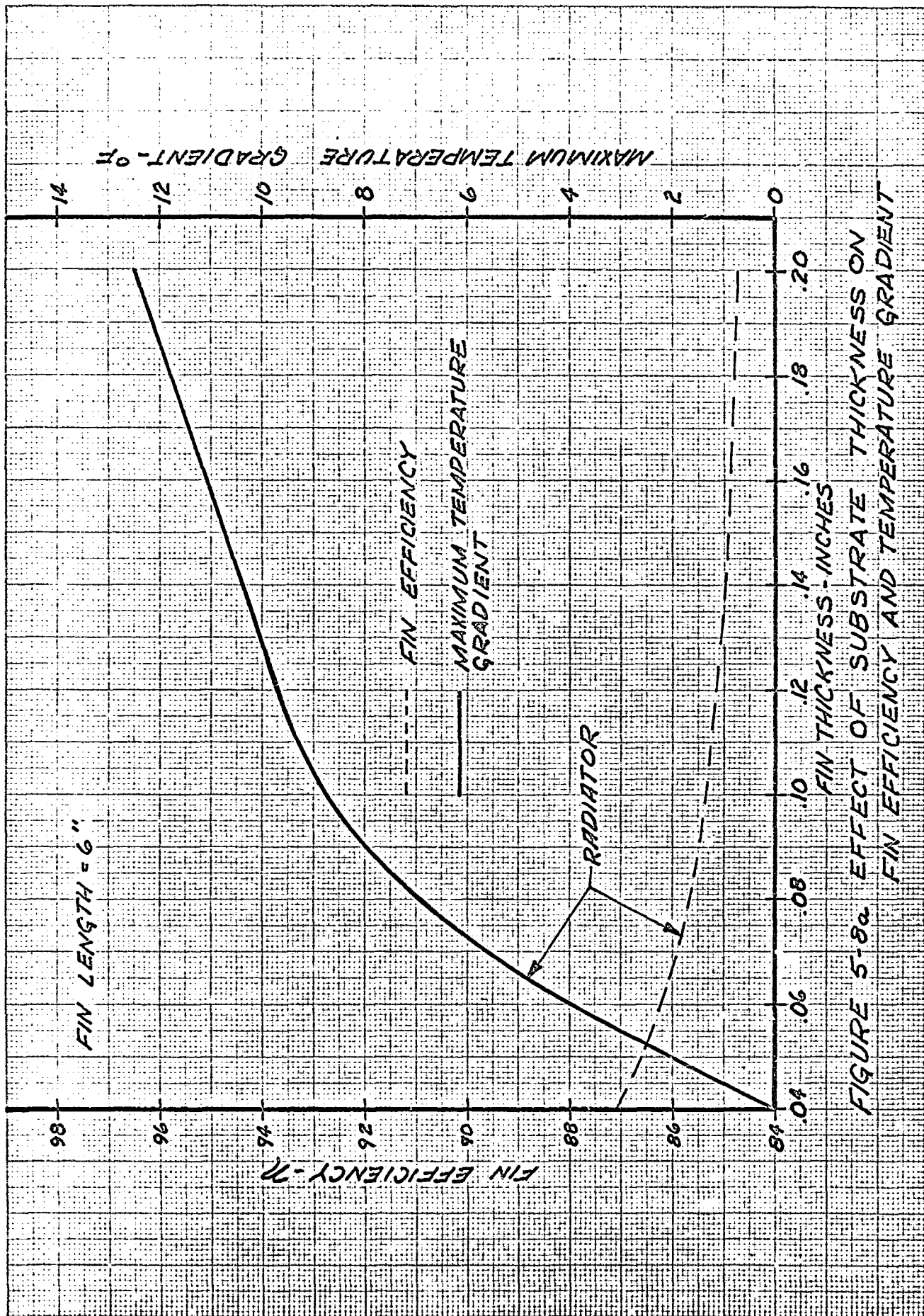


FIGURE 5-8a EFFECT OF SUBSTRATE THICKNESS ON FIN EFFICIENCY AND TEMPERATURE GRADIENT

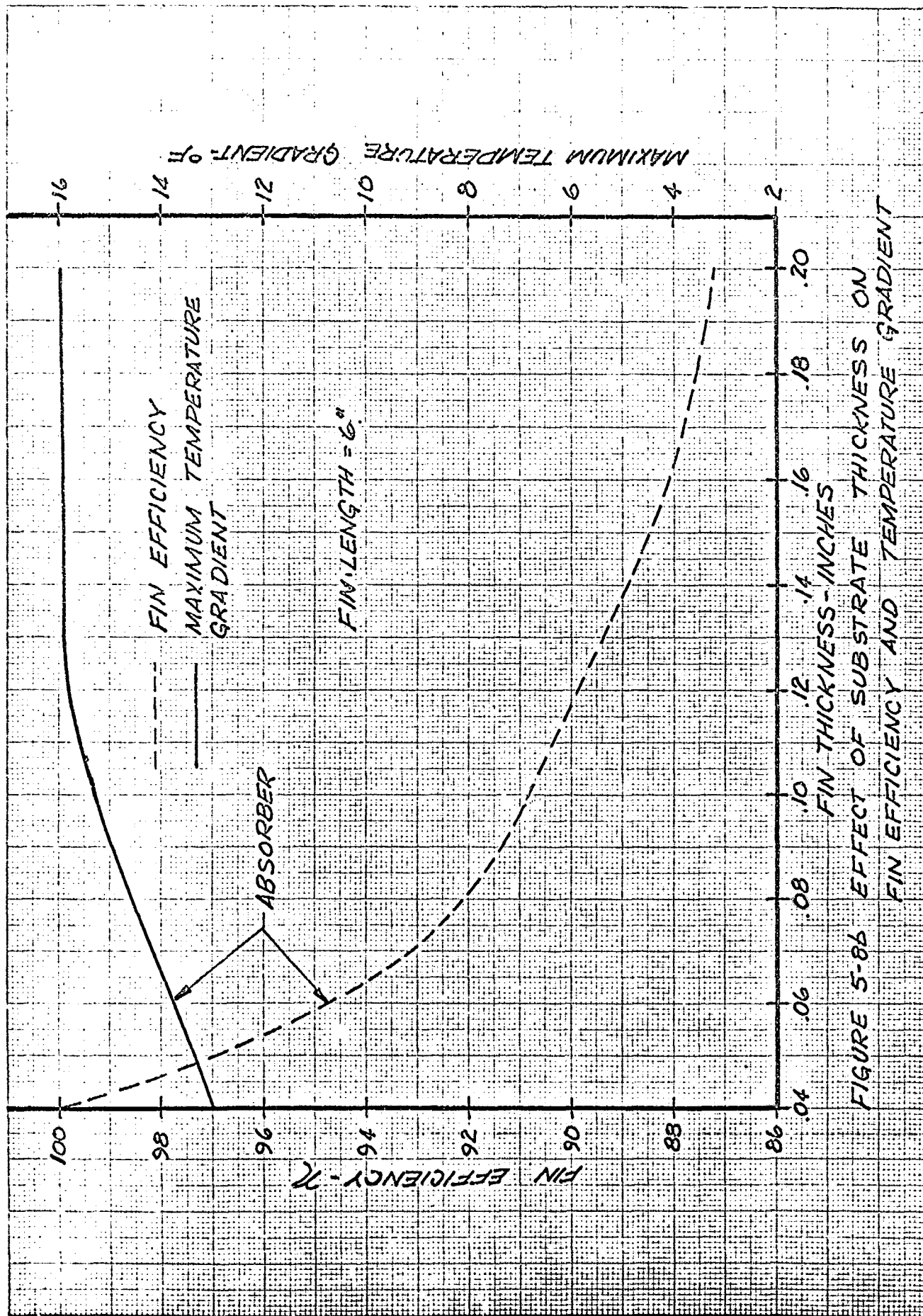


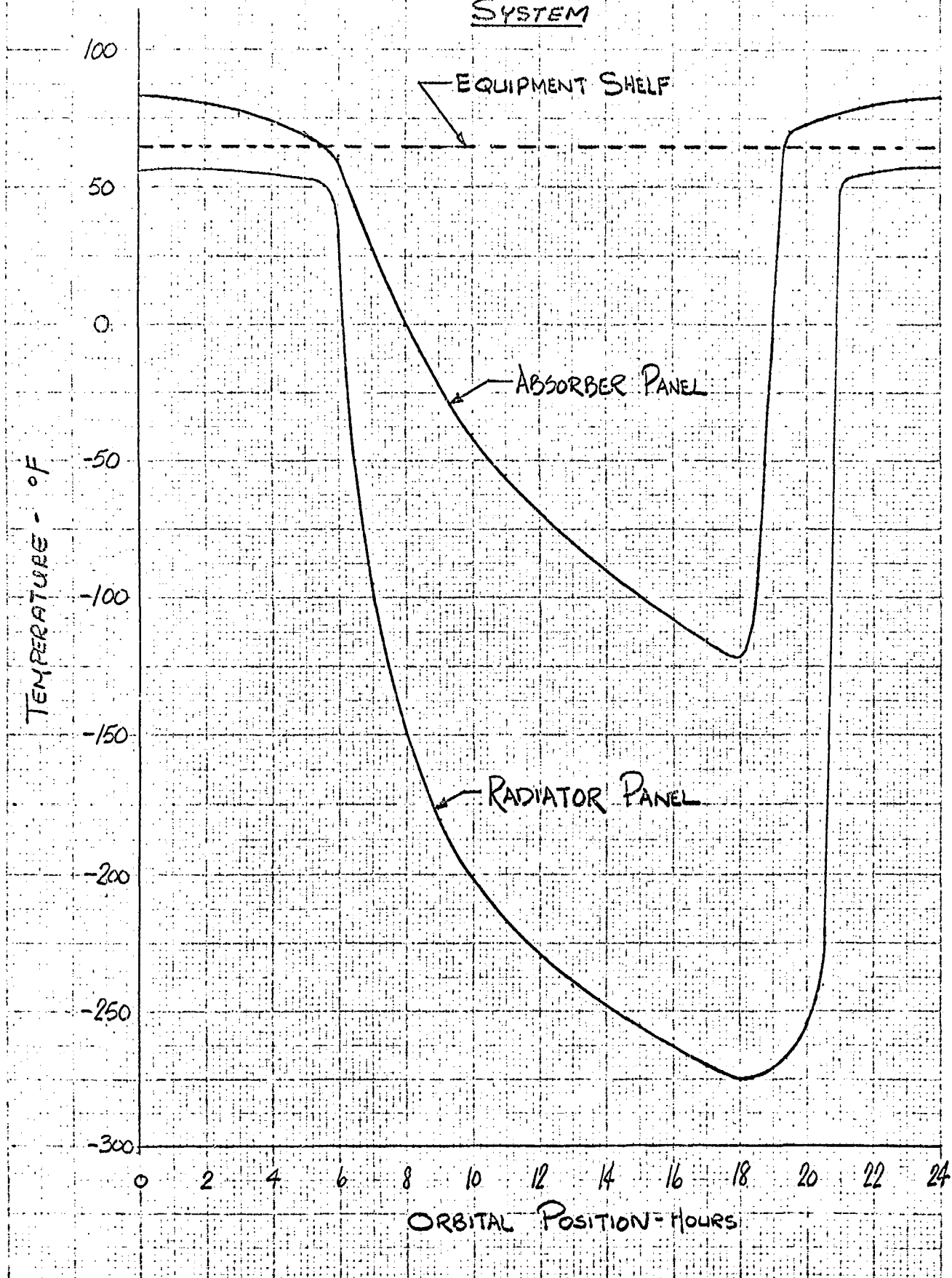
FIGURE 5-86 EFFECT OF SUBSTRATE THICKNESS ON FIN EFFICIENCY AND TEMPERATURE GRADIENT

2. Transient Analysis

During its flight lifetime, the ATFE will experience a nominal daily solar flux profile (Figure 5-3) which will cause cyclic temperature variation of the various components. Transient temperature profiles of the equipment shelf, absorber, and radiator have been calculated assuming ideal operation of the various components. The results are given as a function of orbital position in Figure 5-9.

In summary, the results emphasize that during the shadow portion of the orbit, the absorber and radiator panels are decoupled from the shelf. At this time, the fusible material begins to solidify in order to keep the temperature of the equipment shelf essentially constant and also to maintain a vapor-gas interface in the variable conductance heat pipe. As soon as the absorber panel becomes illuminated its temperature increases rapidly. When the absorber temperature exceeds the shelf temperature the diode turns on and supplies heat to the equipment shelf. After all the fusible material has melted, the variable conductance heat pipe becomes operative and the radiator temperature increases rapidly.

FIGURE 5-9
TEMPERATURE PROFILES FOR IDEAL ATFE
SYSTEM



APPENDIX
GOVERNING EQUATIONS FOR A FEEDBACK
CONTROLLED VARIABLE CONDUCTANCE HEAT PIPE

A. The Governing Relations for a feedback controlled variable conductance heat pipe, which is shown schematically in Figure A-1 are:

1. Energy Balance

$$Q = (m c_p)_s \frac{dT_s}{dt} + Q_{HP} \quad (A-1)$$

also for radiative heat transfer

$$Q_{HP} = (m c_p)_{HP} \frac{dT_v}{dt} + \sigma \epsilon (y - L_E) W (T_v^4 - T_o^4) \quad (A-2)$$

For the steady state analysis Eqs. (A-1) and (A-2) combine to give

$$F(Q, T_v, T_o) = Q - \sigma \epsilon (y - L_E) W (T_v^4 - T_o^4) = 0 \quad (A-3)$$

2. Partial volume occupied by the non-condensable

$$G(V_g, V_v, y, \eta) = V_g - \left[V_{g_o} + A_v (L_c - (y - L_E)) + A'_g \eta - V_v \right] = 0 \quad (A-4)$$

with $A'_g = A_g - A_a$ = Effective cross-sectional area of gas storage volume

The parameter η denotes bellows displacement for the passive system, while the parameter V_v denotes the partial volume occupied by the vapor of the working fluid in the inactive part of the condenser or in the storage volume. The vapor is present due to mass diffusion; or, in the case of an active system, it is present in a saturated condition in a wicked storage volume to provide control.

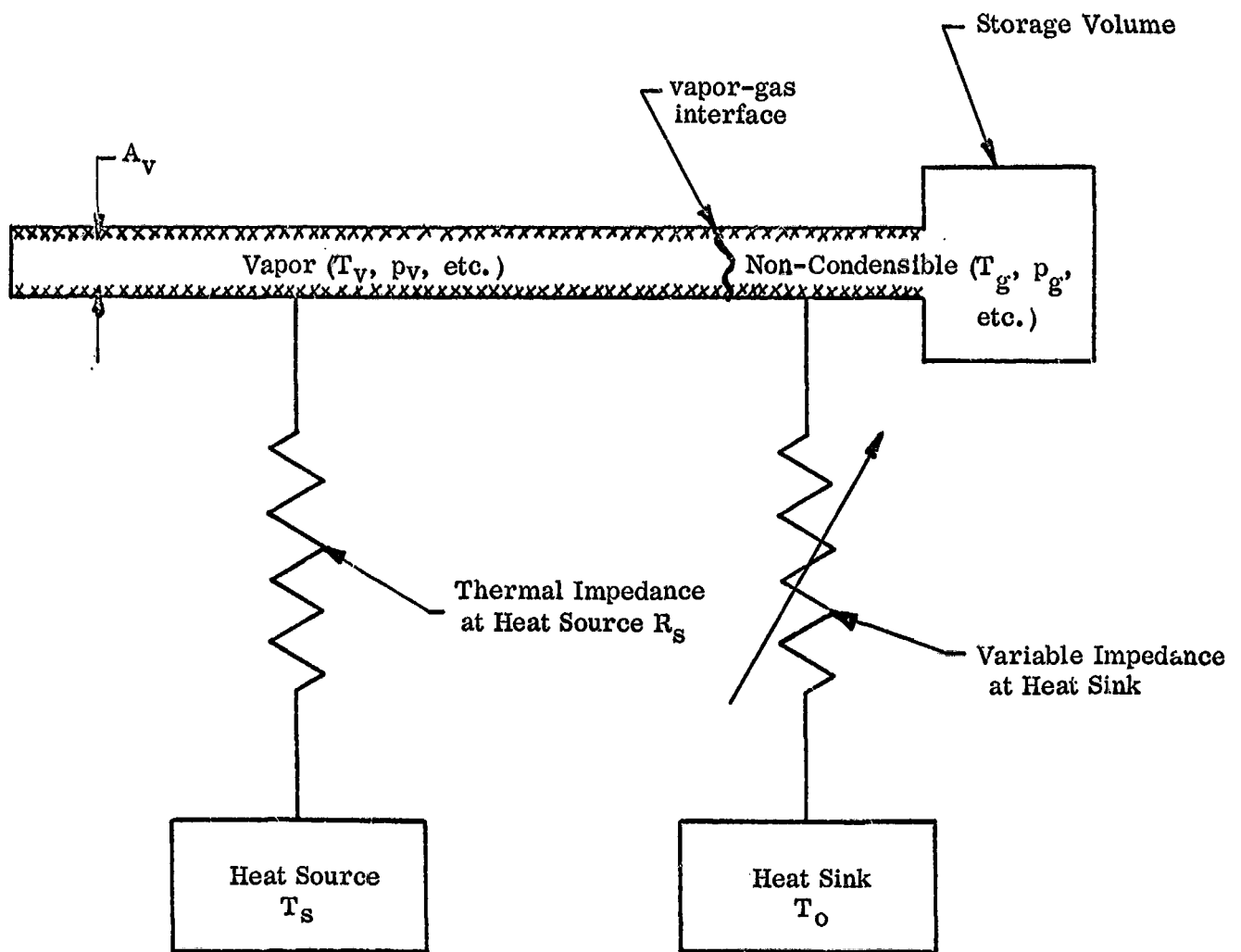


FIGURE A-1
SCHEMATIC OF VARIABLE CONDUCTANCE HEAT PIPE SYSTEM

3. Force Balance (passive system)

For a passive system whose configuration is such that $S_1 > 0$

$$H(\eta, p_a, p_g) = m_B \ddot{\eta} + b \dot{\eta} + k(\eta - \eta_{fl}) - [p_a A_a + p_g A_g'] = 0 \quad (A-5)$$

where η_{fl} = Free length of bellows system

4. Equation of state for the non-condensable

$$p_g (V_g + V_v) = (m R T)_g \quad (A-6)$$

5. Equation of state for the auxiliary liquid sensing the source temperature

$$\frac{\beta}{\chi} (T_s - T_{s_o}) - \frac{A_a}{\chi V_a} (\eta - \eta_o) - (p_a - p_{a_o}) = 0 \quad (A-7)$$

6. Auxiliary Equations

$$\left. \begin{aligned} T_s &= T_v + R_s Q \\ p_g &= p_v - \pi \\ p_v &= p_v(T_v) \\ \pi &= \pi(T_g) \end{aligned} \right\} \quad (A-8)$$

Simultaneous solution of these equations determines the systems performance.

B. The Control Parameters (S_i) are defined in general as follows:

$$S = \Upsilon_v V_g \left(\frac{\partial F}{\partial y} / \frac{\partial F}{\partial T_v} \right) \left(\frac{\partial G}{\partial V_g} / \frac{\partial G}{\partial y} \right) \quad (B-1)$$

$$S_1 = \frac{p_v \Upsilon_v}{1 + \beta_2 \left(\frac{\partial H}{\partial p_a} / \frac{\partial H}{\partial \eta} \right)} \left(\frac{\partial F}{\partial y} / \frac{\partial F}{\partial T_v} \right) \left(\frac{\partial G}{\partial \eta} / \frac{\partial G}{\partial y} \right) \left(\frac{\partial H}{\partial p_v} / \frac{\partial H}{\partial \eta} \right) \quad (B-2)$$

$$S_2 = \frac{\beta_1}{1 + \beta_2 \left(\frac{\partial H}{\partial p_a} / \frac{\partial H}{\partial \eta} \right)} \left(\frac{\partial F}{\partial y} / \frac{\partial F}{\partial T_v} \right) \left(\frac{\partial G}{\partial \eta} / \frac{\partial G}{\partial y} \right) \left(\frac{\partial H}{\partial p_a} / \frac{\partial H}{\partial \eta} \right) \quad (B-3)$$

When a compressible liquid is used as the auxiliary fluid

$$\beta_1 = \frac{\beta}{K} \quad \beta_2 = \frac{-A_2}{K V_2} \quad (B-4)$$

Thus for the configuration shown in Figure 3-1

$$S = \frac{\Upsilon_v V_g}{4 A_v} \left(\frac{(T_v^4 - T_o^4)}{T_v^3 (y - L_E)} \right) \quad (B-5)$$

$$S_1 = \frac{\Upsilon_v p_v}{1 + \frac{A_2^2}{k K V_2}} \left[\frac{(T_v^4 - T_o^4)}{T_v^3 (y - L_E)} \right] \frac{A_g^2}{4 A_v k} \quad (B-6)$$

$$S_2 = \frac{(\beta/K)}{1 + \frac{A_2^2}{k K V_2}} \left[\frac{(T_v^4 - T_o^4)}{T_v^3 (y - L_E)} \right] \frac{A_g A_2}{4 A_v k} \quad (B-7)$$

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